# AN IMPROVED MEANS FOR TREATMENT OF THE GASES OF COMBUSTION ENGINES AND THE TRANSMISSION OF THEIR POWER

Application filed in the U.S. Patent Office
by Mitja V. Hinderks of 1015 Gayley Avenue,
Number 1228, Los Angeles, CA 90024

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## ABSTRACT OF THE DISCLOSURE

The invention relates to a thermally, and optionally catalytically, operative exhaust gas treatment reactor of an engine, comprising filamentary material, directly against and inside the heat insulating portion of a housing assembly which is placed on an engine over the exhaust ports, either directly or with a member interposed. Rapid warm up during cold start is achieved by restriction of gas flow, with the optional re-routing of exhaust gas, including routing to a reservoir which may be expansible. Other features include various embodiments of filamentary material mechanisms for accurately regulating exhaust gas recirculation and provision of extra air, providing a secondary substance such as water/methanol or steam to induction charge to assist in the balancing of exhaust reactions, alternative fuel delivery devices. Treatment of exhaust gas is generally but not specifically to remove certain undesired pollutants from the exhaust gas of vehicles as required in many countries.

## BACKGROUND OF THE INVENTION

The invention relates to a particular method of purifying the exhaust gases of internal combustion engines, and to the benefits to engine construction, fuel conservation and power output that may arise out of the employment of these methods.

Because of the complexity of both the art and the scope of the method of exhaust emission control described, it is proposed to present this disclosure in several distinct sections comprising a preferred embodiment, port area configuration, filamentary material, cold start and related features, reaction process, form of reactor housing, and materials and manufacturing methods generally.

Since all the materials used for any portion of the assembly must have certain features in common, such as heat and shock tolerance, abrasion and corrosion resistance, they have been described under a separate heading and not in relation to individual features, with the exception of certain isolated cases. The brief description of the drawings is continuous, but the discussion has been separated to the relevant sections. The reasons for these arrangements are both to make the location and cross-referencing of information easier and to help the reader to a clearer understanding of the invention.

It is well known that the art of cleaning exhaust gases (as opposed to the art of minimizing the formation of pollutants at the point of combustion) is centered around the technique of speeding up chemical reactions normally tending to continue in the exhaust gases at a slow rate, and that this speeding of chemical reaction is achieved by some combination of two basic means, namely the provision of catalytic agents and the encouragement of reaction under conditions of heat and/or pressure. The provision of extra air is often used to balance properly the chemical reaction to a desired configuration. These methods all involve a majority of common fixtures, such as the employment of reaction chambers, the use of high temperature materials, such as ceramics, the provision of extra air, etc.

The object of the invention is to provide a means of removing noxious or undesirable elements from the exhaust gases from internal combustion, or other, engines, especially in the case of engines fitted in vehicles. The informed reader will

no doubt be sufficiently aware of the general background to the desirability of cleaning exhaust gases and the history and progress of enforcing legislation, so obviating the need to explain these matters in detail here. Any knowledge of the matter must include an appreciation of the difficulties that have been encountered over the years, alluded to above, but were generally caused by the very sudden and great upsurge of activity in the field of emissions treatment, due to the promulgation of legislation initiated by the State of California and later the U.S. Federal Government. The suddenness (and often reluctance) with which the motor industry entered this field resulted in much systemless and uncoordinated work, in which existing techniques and hardware were adapted to meet requirements rather than design approaches being conceived afresh. It is therefore a further object of the invention to overcome the many difficulties and penalties so far encountered with the current means of treating exhaust gases, which are described in greater detail below. The first generation of exhaust emission devices are being fitted to vehicles, for despite their disadvantages the emission laws must be met. It is felt that the present invention constitutes a very much improved second generation approach to the treatment of exhaust gases, for reasons explained in the disclosure.

In general, the problems of early emission systems may be classified under the following five headings, of which the last is perhaps the most important: Adaptability to be mounted on existing engines, ability to meet emission requirements while remaining durable, flexibility of design, cost of installation and maintenance, and cost of operating the engine, and therefore, the vehicle in which it might be installed. Considering the first question of adaptability to be mounted on existing engines, the known systems which offer the promise of good performance are all either very elaborate, involving the use of usually two separate reactors, air pump, exhaust gas recirculation (EGR) sensors, by-pass systems, etc., or they involve different engine configurations or design, as for example rotary, diesel and statisfied charge engines. Under the latter circumstances, it is considered unreasonably expensive in terms of cost

and effort to change the entire engine system, while the question of adapting an existing engine by fitment of existing technology involves such an increased volume of components that the adaption can rarely be made without extensive body and layout modifications. The cost of installing the complex equipment outside assembly-line situations would be very great. As will be seen, the present invention effectively eliminates the need for ancillary equipment (if some is needed for the highest emission standards, a small sacrifice of performance would presumably prove acceptable for existing vehicles). In addition it is a simple device occupying no additional under-hood space, being substitutable for the existing exhaust manifold in a quick and inexpensive operation.

Making all vehicles sold or manufactured in a country conform to a common emissions requirement has generally been a slow and costly procedure. The present invention by its simplicity reduces the variables in practice to the one basic question of type and nature of core configuration. The housing involves no major problems or need for experimentation, and in use it is intended that common housings be adapted to differing engines of similar size or capacity, with core composition modified for the respective engines. Alternatively, a vehicle with the invention fitted could when exported be adapted for the requirements for different countries by substitution of core, without any modification to under-hood or body layout.

The applicant feels that in the long term all emission treatment means will be thermally operative, rather than almost wholly catalytic as the majority of systems today, and this for reasons connected with hardware costs. Basically both heat or catalysts have been employed to achieve the same effect, that is to hasten reaction process. Catalysts are expensive to produce, need costly replacement or maintenance, while heat is available at no cost, since it has already been produced by the process of internal combustion. Therefore economic pressures will ensure that eventually gas treatment largely employs this readily available heat, and the consequent reduced cost

of clean air will make its enforcement practical in areas of the world where it is today considered an uneconomic luxury. To properly use this heat, and thereby reduce ultimate cost, has been one of the prime objectives of this invention. In addition to providing a system wherein catalysts have reduced power (and therefore cost), the invention allows for further economy and maintenance in a number of ways. The present invention comprises a single reactor, operative in the tri-component mode, with further cost savings. Some embodiments involve the use of a single integral housing of high alumina ceramic, which can be manufactured relatively cheaply.

It is hoped that this background note, together with the detailed description which follows will enable the reader to understand fully the objects and advantages of the invention.

## SUMMARY OF INVENTION

The invention comprises an exhaust gas treatment reactor assembly comprising a housing enclosing reaction volume suitable for the passage of exhaust gas, said volume being partly occupied by filamentary material, said housing comprising at least heat insulating material, said insulating material being disposed in the housing most inwardly. The invention further comprises an engine having an inlet system and exhaust port, said port communicating with exhaust gas treatment reactor assembly comprising a housing partly defining a reaction volume, said housing needing to be affixed to the engine in order to suitably enclose said volume and thereby render capable of operation the reactor assembly, wherein an intermember is disposed between said housing and said engine. The invention further comprises an exhaust gas treatment reactor assembly at least partly comprising a casing having a peripheral wall of elliptical configuration in cross-section, the wall of said casing, in plan view, having curved sides gradually narrowing to a blunt apex which forms the exhaust gas discharge aperture.

The invention further comprises an engine having an inlet system, an exhaust port and an exhaust system, said exhaust system communicating with an exhaust gas treatment reactor assembly having gas entry and gas exit, the foregoing components being so arranged that when the engine is operative exhaust gas will pass in a substantially unidirectional manner from a point within said exhaust port to a point beyond said reactor gas exit. The invention further comprises filamentary material suitable for placement in an exhaust gas treatment reactor, said material comprising a multiplicity of pellets, at least one of said pellets having approximately spherical outline and consisting in a series of members substantially projecting from a core. The invention further comprises filamentary material suitable for placement in an exhaust gas treatment reactor, said material comprising a multiplicity of pellets, at least one of said pellets having a surface of approximately spherical outlines, said surface to have at least one substantial depression. The invention further comprises an engine having inlet system, exhaust system, and when operative exhaust gas flow, said exhaust system communicating with an exhaust gas treatment reactor having gas entry and gas exit, said reactor being effectively warmed on cold engine start by means inhibiting said exhaust gas flow, wherein the said exhaust gas flow is at least partly divertable from normal flow down said exhaust system to an exhaust gas recirculation system, wherein said exhaust gas recirculation system communicates with an exhaust gas reservoir. The invention further comprises a valve assembly suitable to be mounted in the fluid flow pertaining to the working of an engine, said assembly comprising a projecting housing within which is a passage communicating with a valve member comprising a shaft attached to wings projecting into said fluid flow, said shaft being slidably mounted and biased by spring action from an open position, to a closed position wherein said passage is restricted. The invention further comprises an engine having an inlet system and an exhaust system, said exhaust system communicating with an exhaust gas treatment reactor assembly, said inlet system communicating via first passage to a chamber in close proximity to said reactor assembly, said chamber communicating via second passage to a fluid reservoir. The invention further comprises an injector

assembly suitable for the injection of fluid into the inlet system of an engine, said assembly comprising a nozzle capable of rotational movement about its axis, said movement being at least partly concurrent with the injection of fluid. The invention further comprises a single injector assembly suitable for the injection of multiple differing fluids into the inlet system of an engine. The invention further comprises a single float chamber assembly suitable to be mounted in association with, and for the purpose of supplying liquid to, the inlet system of an engine, said assembly being capable of simultaneously containing multiple differing fluids. The invention further comprises an uncooled or less cooled engine capable of continuous operation. The invention further comprises an engine operating at least partly on the internal combustion cycle, and having an exhaust system communicating with an exhaust gas treatment reactor assembly, said reactor assembly having disposed within it means defining separate but interconnected volume communicating exteriorly of the reactor, said means being hereinafter referred to as a heat exchanger system. The invention further comprises a multiple apertured hollow needle suitable for placement in inlet system of an engine for purpose of providing fluid to charge. The invention further consists of engine with an exhaust gas treatment system mounted between exhaust port and a turbo charger.

## BRIEF DESCRIPTION OF THE DRAWINGS

A basic embodiment of the principles of the invention in the form of an exhaust gas thermal/catalytic reactor is described below, together with a description of the mode of the operation of the reactor.

In the accompanying drawings:

Figure 1 is a diagrammatic plan view, with a portion removed to show the interior of a reactor assembly according to the present invention.

Figure 2 is a cross-sectional view taken on the line 2 - 2 of Fig. 1.

Figure 3 is a cross section view taken on the line 3 - 3 of Fig. 1.

Figure 4 is a cross section view, similar to Fig. 3, but showing a modified construction.

Figure 5 is a cross sectional view, also similar to Fig. 3, but showing a further modified construction.

Figures 6 to 11 show diagrammatically in vertical cross-section various arrangements of intermembers.

Figures 12 to 14 show in cross-section various fixing details.

Figures 15 and 16 show diagrammatically in sectional plan view two examples wherein reaction volume projects into space normally occupied by the engine.

Figures 17 and 18 show arrangements of exhaust port axes.

Figures 19 to 24 describe means of directing exhaust gas flow.

Figures 25 to 28 describe means of imparting swirl to exhaust gases.

Figure 29 illustrates a preferred embodiment.

Figures 30 and 31 describe honeycomb and wool construction.

Figures 32 and 33 describe expanded metal or metal mesh construction.

Figure 34 describes woven and knitted wire.

Figures 35 to 37 describe wire spiral construction.

Figures 38 to 46 describe wire looped construction.

Figures 47 to 51 describe wire strand and associated features.

Figures 52 to 60 describe various slab-like sheet configurations.

Figures 61 to 65 describe sheet used in three dimensional forms.

Figures 66 to 72 describe details of fixing filamentary matter to reactor housing.

Figures 73 and 74 show an embodiment of exhaust gas reservoir.

Figures 75 and 76 show diagrammatically valve, gas routing and component arrangement.

Figures 77 to 81 show an embodiment of butterfly valve in the situation of Fig. 75.

Figures 82 and 83 show an embodiment of butterfly valve in the situation of Fig. 76.

Figures 84 and 85 show an embodiment of ball valve in the situation of Fig. 76.

Figures 86 to 88 describe examples of valve actuating means.

Figures 89 to 94 describe means of controlling exhaust gas recirculation (EGR) and air supply.

Figures 95 and 96 show embodiments of reservoirs containing multiple substances.

Figures 97 to 99a show embodiments of composite injectors supplying multiple substances to the combustion volume.

Figure 100 illustrates the principle of reduced resistance to gas flow adjacent reactor housing.

Figures 101 to 106 describe configurations of reactor wall construction embodying depressions or projections.

Figures 107 and 108 show a reactor housing and inlet housing assembly, constituting an integral assembly.

Figure 109 shows a component fixing detail.

Figures 110 to 113 show form of reactor housing suitable to V-configuration engines.

Figures 114 to 121 describe means of heat treatment of substances, such as fuel, involved in combustion process.

Figure 122 shows a reactor divided into sections.

Figure 123 shows diagrammatically a way of manufacturing fibers.

Figure 124 shows an isostactive pressing means.

Figures 125 to 131 illustrate pellet-like filamentary material.

Figures 132 to 134, and 183 to 188 show a configuration and details of an uncooled engine or engines.

Figure 135 shows the deployment of heat exchange means within a reactor.

Figure 136 illustrates the interconnection of two engines.

Figures 137 to 139 illustrate linking of crankshaft sections.

Figure 140 illustrates configuration of composite engine.

Figures 141 and 142 show diagrammatically how two engine cycles may be operative on one piston and chamber assembly.

Figure 143 illustrates schematically heat exchange means associated with a reactor incorporated in a turbine engine assembly.

Figure 144 illustrates an embodiment of fluid reservoir of variable volume.

Figures 145 to 149 show embodiments of inlet housings.

Figures 150 to 151 show a combined inlet and exhaust housing assembly.

Figures 152 and 154 show schematically a cylinder head permitting modified port relationships.

Figures 155 to 157 show a variable diameter inlet throat.

Figures 158 and 159 show embodiments of inlet valve.

Figures 160 to 161a show schematically vehicle engine compartment arrangements.

Figures 162 to 165 show schematically embodiments of housing and rotor assemblies.

Figures 166 and 167 show schematically injectors capable of motion in three dimensions.

Figures 168 to 179 show embodiments of movable injectors and/or their locations.

Figures 180 to 182 show schematically heat exchangers associated with turbine assemblies.

Figures 183 to 188 illustrate schematically aspects of uncooled engine construction.

Figure 189 shows the working principles of a tensile link engine.

Figures 190 to 192 show three schematic sections through a five cylinder engine.

Figures 193 and 194 show two schematic sections through a ten cylinder engine.

Figures 195 and 196 show two schematic sections through an eighteen cylinder engine.

Figures 197 and 198 show two schematic sections through a forty-two cylinder engine.

Figure 199 shows schematically a two cylinder engine having connecting rods operating on a single crankshaft.

Figures 200 to 203 illustrate the effects of the effective variable length of the tensile link.

Figure 204 illustrates how both two- and four-stroke principles can be embodied.

Figure 205 illustrates the effect of offsetting the crankshaft axis.

Figures 206 to 209 show details of crankshaft construction.

Figures 210 to 213 show construction details of a first tensile link embodiment.

Figures 214 to 223 show construction details of alternative link embodiments.

Figures 224 and 225 show an interface between tensile link and cylinder head.

Figures 226a to 228 show arrangements of ring valves.

Figures 229 to 235 show schematically methods of fuel delivery.

Figures 236 to 238 show a first embodiment of a piston and cylinder assembly.

Figure 239 shows a method of reducing piston blow-by.

Figures 240 and 241 show a pull-wire valve actuation method.

Figure 242 shows schematically the layout of a two-stroke tensile link engine.

Figure 243 shows schematically the layout of a single crankshaft tensile link engine.

Figures 244 and 245 show diagrammatically a multiple crankshaft tensile link engine.

Figures 246 to 250 show bearing construction details.

Figures 251 and 252 show schematically twin exhaust system engine layout.

Figures 253 to 256 show constructional details of a twin exhaust system engine.

Figures 257 shows schematically a multiple crankshaft engine mounted in a helicopter.

Figure 258 shows schematically a multiple crankshaft engine mounted on a torpedo.

Figure 259 to 261 show schematically a multiple crankshaft engine mounted on a single-pod hydrofoil.

Figures 262 and 263 show engines mounted on a multiple pod hydrofoil.

Figures 264 and 265 show a bearing construction detail.

Figure 266 illustrates the working principle of a continuously variable transmission (CVT) system using belts and rollers of variable diameter.

Figures 267 to 275 show various layouts of the CVT system.

Figures 276 and 277 show an embodiment of a variable diameter roller.

Figure 278 illustrates the relationship between two rollers.

Figures 279 to 289 show details of a first roller embodiment.

Figures 290 to 293 show principles of a second roller embodiment.

Figures 294 to 299 show camshafts and associated followers wherein the camshafts are capable of movement in three dimensions.

Figures 300 to 303 show various three dimensional cams.

Figure 304 shows a cam axial motion actuation device.

Figures 305 to 309 show devices for driving camshafts capable of axial motion.

Figures 310 and 311 show a two-stroke engine capable of having three dimensional cam motion.

Figure 312 shows the cam actuation and timing of a suitable engine.

Figures 313 to 315 show roller type cam followers.

Figure 316 shows a double actuation cam.

Figures 317 and 318 show a ball and sled type cam follower.

Figures 319 to 322 show various rollers.

Figures 323 to 325 show a sled and rocker type cam follower.

Figures 326 to 330 show various cam and follower relationships.

Figure 331 shows a cam follower lubrication system.

Figures 332 to 344 show various follower contact area configurations.

Figures 345 to 347 show schematically a variable lift combined crank- and cam-shaft.

Figures 348 to 350 show methods of varying bearing fluid pressure.

Figures 351 to 353d illustrate the features of torroidal combustion chambers, relative to conventional chambers.

Figures 354 to 359 show schematic layouts of torroidal combustion chamber engines.

Figures 360 to 362 show methods of compensating for differential crank movement in twin crankshaft engines.

Figure 363 shows schematically a one-sided torroidal combustion chamber engine.

Figures 364 to 368 illustrate the principles of imparting three dimensional motion to a piston.

Figures 369 and 370 show schematic layouts of three dimensional motion engines.

Figures 371 to 376 show devices for transferring power from a three dimensional motion engine to a conventional rotating shaft.

Figures 377 to 380 illustrate alternate methods of fuel delivery.

Figures 381 to 386 show engines wherein work is transferred from piston to crankshaft via a yoke system.

Figures 387 to 389 illustrate layouts wherein various working elements are arranged about a common shaft.

Figures 390 to 393 illustrate the principles of sinusoidal torroidal combustion chambers.

Figure 394 illustrates a two-stage torroidal combustion chamber engine.

Figure 395 and 396 show part profiles of sinusoidal torroidal combustion chambers.

Figures 397 to 399 illustrate the principles of the sinusoidal torroidal combustion chamber engine/differential.

Figures 400a to 404 show constructional details of sinusoidal torroidal combustion chamber engines.

Figure 405 shows schematically an engine with multiple sinusoidal torroidal combustion chambers.

Figures 406 to 407 illustrate methods for varying ratio of rotational to reciprocating motion.

Figures 408 to 411 show alternate gas flow arrangements.

Figure 412 shows a combustion chamber profile.

Figures 413 and 414 show schematically engines or pumps having two working chambers, one torroidal, one conventional.

Figures 415 to 425 illustrate the principles and constructional details of modular engines.

Figures 426 to 430 illustrate schematically various post arrangements for hydrofoil craft.

Figures 431 to 437 show elevationally various configurations of keel elements.

Figures 438 and 439 show an extensible hydrofoil.

Figures 440 to 443 show extensible hydrofoil constructional details.

Figures 444a to 450c show various post, keel and hydrofoil assemblies.

Figures 451 to 456 show retractable keel assemblies and their relationship to vessel hulls.

Figures 457 to 459 show schematic plan, elevation and multiple section of a vessel with retractable keel assemblies.

Figures 460 and 461 show an underwater exhaust discharge assembly.

Figure 462 shows a marine engine pod with underwater exhaust discharge.

Figure 463 shows a marine propeller drive with coaxial exhaust discharge.

Figure 464 sows exhaust discharge at marine propeller surfaces.

## DESCRIPTION OF THE PREFERRED EMBODIMENTS

In carrying the invention into effect as shown by way of example in Figures 1 to 3, the reactor assembly comprises an outer metal casing or chamber 10, an inner casing or chamber 11 of solid ceramic material conforming in shape to the inner surface of the outer casing 10 and a layer of fibrous material 12 interposed between the inner and outer casings. The periphery of both the outer casing 10 and layer of fibrous material 12 are provided, respectively, with flanges 13, 14 having a plurality of aligned apertures through which bolts 15 pass to mount the reactor assembly on an engine block 16 so that all the exhaust ports 17 of the block communicate with the

interior of the inner ceramic casing 11. Filamentary material such as nickel chrome alloy is accommodated in the inner casing 11 in two forms, i.e., firstly randomly disposed wire 18 and secondly a spiral coil 19 of thicker wire mounted adjacent each exhaust port 17 in order to reduce the velocity of the exhaust gases at the port outlet.

It will be useful here to summarize the working principles of the reactor prior to the full description following later in this section. In operation, due to the positioning of the reactor on the engine and the insulation of the inner surface, the contents of the chamber, i.e., gases and filamentary material, are maintained at a high temperature, so that the exhaust gases discharged from the engine cylinders continue to oxidize and react after entering the ceramic casing 11, thus substantially eliminating unburned hydrocarbons, carbon monoxide, and the oxides of nitrogen of the exhaust gases. In addition, the filamentary material 18 acts as a filter to trap any solid particles in the exhaust gas and induces localized turbulence which pushes the maximum quantity of gas into contact with the hot surfaces of the filamentary material in the shortest possible time.

In order to ensure rapid warm-up of the filamentary material 18 and 19 during cold starting, a valve member 20 is pivotally mounted on a spindle 21 adjacent the discharge end of the reactor assembly, the metal casing 10 and layer of fibrous material 23 of which are provided, respectively, with flanges 22 and 23 which, as shown in Figure 3, are connected by bolts 24 and retaining nuts 25 to the flange 26 of an exhaust pipe 27 forming part of the exhaust system of the vehicle. Under cold starting conditions, the valve 20 is closed either manually or automatically (generally two or three cycles after firing commences) by linkage 28 so that the newly fired exhaust gases are retained in the chamber 11 to ensure a rapid temperature rise therein until a predetermined pressure is reached, whereupon the valve member 20 is opened, at least partially. Conveniently, this may be effected by having the valve 20 biased to a closed position by a torsion spring (not shown), operative only during the cold start procedure,

and mounted on a spindle 21 which is diametrically displaced so that the increased pressure in the reactor assembly applies a turning moment to the valve member 20 which commences to open when the moment exceeds the closing force exerted by the spring. A pressure relief valve 40 and passage 41, shown diagrammatically in Figure 1, may be provided in the chamber anterior to the valve member 20.

It will therefore be appreciated that the normal position of the valve at the discharge end of the reactor retains the exhaust gases in the chamber with a consequent rapid rise in the temperature of the filamentary material, which in turn assists in the continued reaction of the trapped gases. A similar, although less intensive, effect is achieved by the partial closure of the valve member, which by the build up of pressure delays the normal passage of the exhaust gases, which thereby remain longer in contact with the filamentary material and heated surfaces and are encouraged to react, e.g., by oxidation and/or reduction.

The modified arrangement shown in Figure 4 is suitable for use with a high performance engine where maximum insulation may not be desired and the firm mounting of filamentary material may be important. In this embodiment one end of the spiral coil 29 which has a thickened externally threaded base is screwed directly in the exhaust port 17 which increases heat transfer from the outgoing exhaust gases to the surrounding block 16 and engine cooling systems. The chamber housing shown partly in section at 42 illustrates an alternative construction comprising integral ceramic shell held in position by "L" clamps 43 and bolts 15.

In the modification shown in Figure 5, if it is found necessary to reduce heat transfer from the outgoing exhaust gases to the surrounding block 16 and cooling system, each port 17 is provided with a sleeve 30 of ceramic material which has a layer of fibrous material 31 interposed between its outer surface and the block 16. A skin 32 of metal or other material is shown placed within the insulation in order to assist in

the reaction process. In Figure 5, it is shown diagrammatically, but in a preferred embodiment, this skin of metal or other material is of no significant thickness and constitutes a film which has been applied by a deposition process, or a leaf (say of similar configuration to gold leaf) applied by pressure and/or adhesive. The film may further be applied to a say ceramic structure by means of depositing the material in metal powder form on the surface of a mould during the process of manufacturing such ceramic structure. Where this process entails forming under heat and/or pressure, the foreign material will be bonded to the surface of the ceramic to substantially form a film.

Catalysts may be associated with the reactor assembly to assist in the removal or transformation of the undesirable constituents in the exhaust gases. The embodiment described above relating to metal or other films describes how a catalyst may be associated with the internal surface of the reactor, but to be properly effective the catalyst should be present throughout the chamber, so that all the gases may be exposed to catalytic action. Catalysts may be incorporated in or with the filamentary material disposed within the chamber. By catalyst is often meant materials with very strong catalytic action such as noble metals like platinum, palladium, etc. However, in this disclosure catalyst is meant to be any material having a significant, measurable catalytic effect and thereby is certainly included materials having only moderate catalytic effect, such as nickel, chrome, nickel/chrome alloys, etc. The conventional approach to the provision of catalytic action within exhaust reactor systems involves the placing of strong catalysts such as noble metals in small quantities on a supportive material such as ceramic. In a similar manner, the filamentary material may have deposited on it small quantities of another material having catalytic properties. Alternatively the filamentary material may be constructed of a material which itself has a moderate to good catalytic effect, such as nickel/chrome alloy.

The filamentary material may consist of high temperature metal alloy, such as stainless steel, Iconel, or ceramic or "plastics" material, i.e., materials of the "giant molecule" family, having molecular weights in the over 5,000 range, including such generic materials as polymers, hydrocarbons, resins, silicons, etc. These are more fully described hereinafter. By the term "filamentary material" is meant portions of interconnected material which allow the passage of the gases therethrough and induce turbulence and mixing by changing the directions of travel of portions of the gas relative to each other. Such material conveniently takes the form of random or regularly disposed fibers, strands or wires, but may also take the form of multi-apertured sheet or slab, cast, pressed or stamped three dimensional members having extended surfaces.

The chamber housing may be constructed as already described, i.e., either from solid ceramic or a multiple layered construction comprising an internal skin of ceramic, an interlayer of fibrous material such as ceramic wool, and an external structural casing of metal. Any suitable high temperature material having good structural and/or insulation characteristics may be employed, including materials of the giant molecule family mentioned above. The housing may be of composite construction, e.g., with one layer manufactured inside or outside of another already manufactured layer. In this way, a layer of high temperature resin, having very good insulation qualities but not very resistant to abrasion or corrosion, may be formed outside of a ceramic shell which, because of its hardness and greater temperature tolerance, will be less resistant to attack by the exhaust gases, as more fully described subsequently.

It will be appreciated that, if desired, provision may be for the entry of additional air into the chamber to assist in the combustion process. As mentioned above, it is felt to be a desirable object to eliminate the need for extra air, and it will be explained later why under normal circumstances the desired reactions may take place without the need for extra air. However, it is envisaged that the invention may

be adapted to existing engines or engine designs, and these may sometimes have special combustion characteristics which require extra air, perhaps under some running conditions. Similarly, some engines are purposely built to operate under very special running conditions, for example, to power heavy earth-moving equipment, and for such applications the provision for extra air may be desirable.

In operation, the device described above will act as a thermal/catalytic exhaust gas reactor, that is to say, it is capable of achieving its objective of hastening the process of reaction by the provision of both a high temperature environment and a catalytic action in the same reactor assembly. For reasons which will be more fully explained later, it is the temperature aspect which is in general more important, i.e., more effective, and the catalytic action can be said to be, in some applications, an assistance to the temperature-oriented process. It is possible, with basically very clean engines, to envisage de-polluting the exhaust gases to the highest standards with negligible or coincident catalytic action. By coincidental is meant that materials having some catalytic effect may be present in contact with the gases for reasons unconnected with catalytic action, that is, they may be the most suitable materials to meet certain design parameters, such as high temperature resistance, etc. The principles of the invention will remain largely the same in the many embodiments suitable to be adapted to all internal combustion engines, and will here be described in general without reference to specific data, which will of necessity only apply to a particular engine. Where applicable such data will be given subsequently. It will be appreciated that engines have widely differing compression ratios, exhaust gas temperatures, gas flow rates, relationships of power to torque, as well as varying operating modes, but the basic principles of the invention will be applicable to nearly all internal combustion engine configurations.

The invention will constitute a very effective thermal reactor. High working temperatures will be attained because of the proximity to the exhaust ports,

which discharge directly into the reaction volume, and its shape which entails a small external surface in relation to volume, so keeping heat loss to a minimum. Conventional thermal reactors, of which perhaps a typical example is described in Behrens U.S. patent 3,247,666. usually involve a plurality of stub exhaust pipes discharged into a narrow cylinder, and here the surface areas are high in relation to reactor volume, with consequent greater heat loss. The conventional configuration also poses problems with insulation, since the most suitable material, ceramic, cannot readily be made reliably in these relatively complex shapes, cracking occurring where one cylindrical shape meets another. The invention, with its flowing rounded shapes, inherently very strong, is more suited to application of considerable insulation, which can readily be manufactured in the most suitable material. The provision of exhaust port insulation, for example as in Figure 5, will further eliminate the heat losses otherwise carried through the block or cylinder head to the cooling system. Because of the shape of the invention. which can broadly be described as a form of inverted megaphone, and the presence of filamentary material (perhaps of a wool-like configuration) internally, it will act to a significant degree as a muffler. It is known that a muffling effect involves dissipation of sound waves, whose energy is converted to heat which remains residual in the muffling agent. In this manner a significant additional build up of heat will occur in the filamentary material and on the walls of the chamber, due to the dissipation of sound waves and also of physical vibration. The main chemical processes, which will be described later. all involve oxidation in part of the reactions, and this produces further considerable heat. It is estimated that because of a combination of all or some of the above factors, ambient temperatures in the invention will be higher than at the exhaust port of an untreated engine. Temperatures drop during idle or low-load conditions, and here the invention will be at an advantage over some other systems, in that a relatively thick ceramic shell will act as a heat sink (as do ceramic linings in many industrial processes) and cause some heat to be radiated inward if the exhaust temperature drops below that of the inside of the housing. This radiation will be directed to maximum advantage because of the rounded or radial cross-sectional form of the housing.

The beneficial effects of the high ambient temperature are most efficiently exploited in the present invention principally through the provision of filamentary material, which, in effect, means exposing the exhaust gas to a multiplicity of hot surfaces. It is known that for some reason, apparently still not fully understood by thermodynamicists, chemical action more readily takes place in the presence of a heated surface. This phenomenon is distinct from catalytic action, which relates to the nature of materials. Therefore the provision of multiple, closely spaced heated surfaces in the form of filamentary material ensures that every portion of the continuously reacted exhaust gases is in close proximity to a heated surface. Further, the exhaust gases are immediately exposed to such surfaces on leaving the port, when they are at their hottest and most ready to react. The filamentary material has the additional advantage of inducing minor turbulence, causing the various portions of the gases to mix properly with each other, thus helping the reaction process and also causing some heat to be generated by the kinetic energy of gas movement. This turbulence is important for another reason in that it allows the composition of the gases more readily to "average out." During the process of combustion, different products are formed in the various portions of the cylinder, due to differences in temperature, the variable nature of flame spread, locality of spark plug and fuel entry, presence of fuel or carbon on the cylinder walls, etc. Usually these differing products of combustion are mixed to some degree in their passage through the port, but nevertheless pockets of a particular "non-average" gas may persist, and these will not have the proper composition to interact in the desired way. This can occasionally present difficulties, for instance in the long unconnected capillary passages of the honeycomb structures used for catalysts, if these are mounted too near the exhaust ports. The nature of the filamentary material of the invention ensures that this proper "averaging out" or intermixing of gas composition takes place. Conventional reactors, as for example by Behrens, are in comparison with the invention relatively crude in this respect. Great columns or cylinders of gas flow through the apparatus, which is only affecting a very thin periphery of the gas.

It is intended that the invention incorporates catalytic agents of whatever nature and strength desired, depending on such factors as the efficiency of the thermally assisted reactions, the type and quantity of pollutants that are needed to be removed, durability, the particular additives of the fuel, etc. There has already been described how coatings of catalytic materials may be applied substantially to the various surfaces of the reactor interior. In a preferred embodiment the filamentary material itself is manufactured from material having catalytic effects, such as nickel, nickel/chrome, copper, stainless steel, etc. Nickel/chrome alloy is a most suitable material, since it is not too expensive and is relatively resistant to corrosion, abrasion and high temperatures, having a moderate to good catalytic rating. However, at the high ambient temperatures of the invention, nickel/chrome will have formed on its surface films of nickel chrome oxide, which has a catalytic rating considerably better than that of its base. Such material, disposed in filamentary form, will have a strong catalytic effect.

Most catalytic activity has involved placing the catalyst relatively far from the exhaust ports where temperatures have been in roughly the 200°C. to 500°C. range, because the noble metal catalysts, or their method of fixing to base material, or the form of the base material (often honeycomb ceramic) has not been reliable or durable at higher temperatures. It is known that catalytic effectiveness can increase logarithmically with temperature increase, in roughly squared proportion. In other words, doubling the temperature can give around four times the effectiveness, tripling the temperature nine times the effectiveness, etc. Of course, this is an extremely rough guide, there being no such clear cut mathematical progression, much depending on materials and circumstances of reaction. For example, certain catalysts become effective within a relatively small temperature increase and then do not greatly increase effectiveness with further substantial rise in temperature. But in general, catalytic effectiveness increases substantially with increase in temperature, as shown in work of G.L. Bauerle, and K. Nobe (among others) in their paper of September 1970 for Project Clean Air, as-

sociated with the University of California. The present invention offers scope for using known catalysts more effectively than ever before, since they will operate in temperatures significantly higher than those currently employed in catalytic practice.

The filamentary material, together with the high ambient temperatures, will ensure that the invention will be exceptionally tolerant of particulate matter and impurities or trace materials, such as for example lead compounds. The filamentary material, especially if at least partly of fibrous or wool-like configuration, will to a great extent act as a trap for particulate matter, without the lodging of such matter in the reactor significantly affecting the latter's performance. Certain other systems, such as catalytic honeycomb structures are notoriously sensitive to particulate clogging, damage by impurities originating in the fuel or by operator misuse. The vast majority of any particulate matter lodged in the present reactor system, with its exceptionally high ambient temperatures, would decompose, oxidize or otherwise react, especially if deposited on surfaces having catalytic characteristics. As with catalytic effectiveness, substances generally increase their tendency to react or combine in logarithmic proportion to temperature increase, in other words a substance will roughly react four times as easily if temperature is doubled. Of course, the provisos mentioned above apply, but, this very broad principle can be said to be the theory behind thermal reactors in general. It is for this reason also that the invention is considered to be primarily a thermally operative reactor, the manner and effects of this thermal action being described above. Although it may also operate very effectively in the catalytic mode, this operation is enhanced and affected by the high working temperatures attainable within the assembly.

Both in its thermal and catalytic operating modes - which in practice merge to form a homogenous encouragement for matter to combine - the reactor is intended to function in the tri-component or three constituent mode, that is the three principal pollutants are all reduced during their passage through the single device. The

three main pollutants are hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NOx). Industry interest in the three constituent approach has only developed significantly since the early seventies, and first mention of successful laboratory results were apparently in a paper presented by T.V. de Palma at the Interpetrol Congress in Rome, Italy, on the 24th of June 1971. This and subsequent efforts have all used catalysts of the conventional configuration (although not necessarily material). By an extraordinary coincidence, calculations and practical results have shown that the three constituents are most likely to be simultaneously reduced at fuel/air mixture ratios of 14.7 to 1, in other words at stoichiometric mixtures, those producing the optimum overall power or work from the combustion of a given amount of fuel. The practical and commercial significance of this is mentioned above, but can be summarized as meaning that no modifications to the combustion process nor additions to the exhaust reaction process need be made. This means that virtually all such power-sapping and/or expensive devices such as extra air, exhaust gas recirculation, ignition timing alterations from optimum, enriching or weakening the mixture, etc., could be eliminated if the tri-component approach to exhaust gas treatment is successfully employed. Briefly and very simply explained, the tri-component process functions in the following manner: Firstly, the hydrocarbons react with oxygen normally present in the exhaust gases, since this is the most readily formed oxidization. Next, the carbon monoxide unites with the remaining oxygen to form carbon dioxide. However, not all the carbon monoxide has been disposed of, approximately half remaining, and this becomes sufficiently "oxygen hungry" to attack the oxides of nitrogen, transforming itself to carbon dioxide, leaving nitrogen and other compounds.

The first results, although successful under laboratory conditions, were impractical commercially because of the extreme sensitivity of the catalyst and excessive degree of mixture control required. Although mixture control techniques have improved greatly in the intervening period and catalysts have become more tolerant, this is still a basic problem with the pure catalytic approach. The present system offers

advantages over prior systems, in that its operation is thermally oriented and it contains filamentary material. The clogging and poisoning of conventional catalysts is no longer a problem, and the invention's tolerance of impurities can be an advantage. The tri-component process depends on critical adjustment of fuel mixtures; when the mixture becomes unbalanced there arises an excess or lack of one of the constituents and so the reactions do not balance out, leaving pollutants still "unreacted." Usually this involves some form of starvation, i.e., there is no material left for a pollutant to react with. The quantities of "unreacted" pollutants are generally very small, and because of the "starvation" situation would have a tendency to react with other matter present. Thus, the present invention, which is impurity tolerant and would have disposed within it significant traces of impurities or secondary material, will tend to be more suited to the tri-component approach than many current systems.

A further very important advantage of the present invention is that it employs the tri-component approach using primarily thermal means, unlike all other tri-component systems known to applicant which all employ a substantially catalytic approach. Of the three principal reactions, that between the carbon monoxide and the oxides of nitrogen is the most difficult to achieve. Traditionally, the removal of the NOx has presented greatest difficulty in nearly all emission systems, basically because of the relative reluctance of NOx to react with other substances. This has meant the industry-wide employment of powerful catalysts and was the impetus behind the employment of expensive noble metals such as palladium, etc. This reaction is also the one needing the most assistance in the tri-component process. However the two chemical types involved here, CO and NOx, change their characteristics markedly with increase in temperature. At the ambient temperatures of the present invention they behave differently than such gases in current catalytic systems, which may be up to 700°C. cooler. The oxides of nitrogen (in fact a family of compounds having broadly similar characteristics) become unstable at high temperature, having the tendency to break into their constituent elements and form new reactions. The carbon monoxide

becomes extremely oxygen hungry with increase in temperature and will far more readily form the desired reaction with the less stable NOx. The effect on chemical characteristics and stability is here again roughly in logarithmical proportion to increase in temperature. In other words, it is one of the prime advantages of the invention that the tri-component process takes place in a high temperature environment. It is felt that this method offers the best long term approach to the treatment of the most difficult pollutant, NOx.

The first attempts to solve the emission problems used a thermal approach, because of its many inherent advantages. Work was gradually abandoned because of the great difficulties of the cold-start situation. To be effective the reactors had to be hot; warm up took a considerable time, during which an unacceptable level of pollutants were emitted.

It was to overcome this traditional problem that the cold-start procedure of the present invention was evolved. A reactor inevitably has a considerable mass, so efforts were made to devise a system whereby at least the effective working parts of the reactor attained the desired temperature, rather than the whole assembly, including parts not affected in the reaction process. The surfaces of the present invention are its effective working parts, and almost wholly comprise the internal lining of the housing, consisting of insulating material, and the internally disposed filamentary matter. The insulating material, such as ceramic, may have a low conductivity and therefore will not significantly transmit heat from the interior of the chamber, nor will it need much heat input to heat the surface molecules to the internal ambient temperature. (Because of low conductivity, the surface molecules do not readily conduct heat to adjacent more inwardly disposed molecules). It is for this very important reason that the invention has its reaction volume directly enclosed by insulating material. The interior filamentary material essentially has low mass and extended surface area (unlike the heavier baffles or internal chambers of some traditional reactors). As will be described more

fully in later sections, the filamentary matter may be of a wide range of materials, including for example metals and ceramics. If metals are used, their conductivity ensures that heat will be absorbed in heating their entire mass, while in the case of ceramics, for the reasons mentioned in connection with the housing, very little heat would be absorbed in bringing surface temperatures to the required levels. It is important to emphasize that the heated surfaces of the reactor are its effective working parts and that therefore only their surfaces need warm up rapidly.

It is in order to use heat already available from the process of combustion (rather than purposely provided for initial cold start) that the gas exit from the chamber is at least in part closed after firing commences. Calculations have shown that, provided all the newly fired gases can be retained by the chamber, its working surfaces will attain temperatures of 700 C. within between about five and fifty cycles after firing commences, depending on engine type, degree of conductivity of the filamentary material, whether exhaust port insulation is fitted, etc. It has been assumed that the total reaction volume is approximately double the engine capacity and that roughly 500 grams of filamentary material are employed for every two liters engine capacity. At idling speeds of 1,200 r.p.m., a four-stroke engine would have, according to the above, a warm up period between half a second and five seconds. A contributing factor to the temperature rise is the fact that the gases are maintained under pressure, this pressure soon contributing some load to the pistons, and thereby enabling the engine and especially the combustion volumes to warm up more rapidly.

In a preferred embodiment, the reactor gas exit is closed in cold start by mechanical or automatic means after firing has commenced and just prior to the newly fired exhaust gases reaching the closure means, which in the case of four-stroke engines will be somewhere between two and five cycles after firing commences, depending on reactor volume, etc. This allows the residual gases to be expelled and ensures that all the thermal energy produced by the combustion process and contained in the exhaust

gases at the ports is entirely used to heat the working surfaces of the invention, and accounts for its rapid warm up. The newly fired trapped gases are reacting in the desired fashion, but more slowly than they would at normal working temperatures. The fact that they remain much longer in contact with reactor surfaces than they do under normal running high temperature situations compensates for their slow reaction rate and ensures that the first gases are largely pollutant free when they leave the reactor. This feature is of great importance where the invention has been fitted to a vehicle seeking to meet for example U.S. emission regulations, important sections of which are enforced through cold-start tests. The requirements of these tests have not always been easily met by other systems, especially some thermal reactors, but the present invention has the unique advantage of producing zero emissions, in fact no exhaust gas whatever, during cold start.

The minimum number of cycles (i.e., firings) needed to reach reactor operating temperature and the maximum number of cycles which may elapse before the exit need be closed are sufficiently near overlap to ensure that the newly fired exhaust gases can be totally contained (i.e., the closure member be totally closed) for at least a substantial, very possibly the whole part of the cold start procedure, depending on such parameters as engine and reaction construction, volume relationships, etc. In a preferred embodiment, the closure member remains wholly closed until a pressure is reached inside the reactor, which is just below that which would cause the engine. which is pumping against reactor pressure, to stall on idling. In use it is preferred that the vehicle be not drivable during the few seconds of the cold start procedure, since pressure below optimum for warm-up procedure must be adopted if allowance is to be made for possible clutch engagement. The reactor pressure limit may be increased by the provision of either manual or automatic special engine setting, such as altered ignition or valve timing, special carburation, alteration of compression ratio, etc., during the cold start procedure. Once the maximum allowable pressure in the reactor has been reached, the gas exit closure member may either (a) wholly open to release

pressure and bring the system to normal running, (b) part open to maintain the pressure, allowing gases to leave the reactor at approximately the same rate as on entry, (c) remain closed while a second closure member wholly or partly opens to relieve or maintain pressure and conduct exhaust gases through a passage other than the normal exhaust system. This alternative is discussed more fully hereinafter. Alternative (b) allows the cold start procedure effectively to continue, since the maintenance of reactor volume pressure ensures that the gases spend longer in their passage through the chamber than under normal running conditions, this lengthening of passage time enabling the gases better to transfer heat to the colder reactor surfaces and to remain in a reacting environment for a more extended period to compensate for colder temperatures, so enabling the anti-pollution reactions substantially to take place. In a similar manner, alternative (c) also allows the cold start procedure to be maintained. In the preferred embodiment the first closure member is fully opened when the desired operational temperature is reached. The resultant pressure drop as normal gas flow rates commence will normally cause an initial surge in engine idling revolutions, giving the operator an audible indication that the engine is ready for work, and the clutch may be engaged.

The invention can be embodied in forms to meet the most stringent emission requirements, and perhaps meet them by wide margins. Considered as a catalytic reactor it can be incorporated in highly effective embodiments. Considered solely as a thermal reactor, it can be embodied to function at least as effectively. The provisions for cold start place the invention at an advantage over competitive systems, mostly emitting a high rate of pollutants during cold running, and taking significantly longer to warm up than the present assembly. A further factor contributing to the effectiveness of the invention is the fact that its volume may be relatively larger than other systems fitted to an equivalent engine or vehicle. This is because the basic form of the invention involves the incorporation into the reaction volume of space not normally considered usable, that is the space in other systems between stub manifolds

and between the connecting manifold or reactor and engine/cylinder block. With basically unidirectional gas flows, an increase in reactor volume retains the gases for longer in the reaction environment, thereby improving the degree of pollutant removal. Because the invention is effective in four separate respects, it offers capacity to produce pollutant levels well below required levels under normal running and cold start conditions. Because the emission requirements in many countries are based on cumulative measurements, that is totals over a spectrum of time and/or operating conditions, this normally below average performance means that it is of less importance if, under certain infrequent conditions or modes of operation, a temporary excess of pollutants is produced. The temporary excess is easily lost in total emission levels which are generally well below requirements. This characteristic is especially useful in reactors operating in the tri-component mode, which is sensitive to great mixture-ratio variations.

Materials and methods of manufacture are described in detail hereinafter. In summary, it is felt that the invention should be mass producible at a cost very much lower than other systems. The housing can be manufactured to last the life of the vehicle, as can the filamentary core if it is of ceramic material, including glass. If of metal such as nickel/chrome alloy, the core could be expected to last a minimum 25,000 miles, being easily and cheaply replaceable.

The above is meant to constitute a simple, easily understandable description of the basic features, principles and advantages of the invention, as it may be embodied to be fitted to any internal combustion engine and is intended to be understandable by persons not normally engaged in the art of exhaust emission control. It is hoped it has been shown that the invention overcomes to a significant degree various difficulties encountered in the art, such problem areas including the questions of cost, fuel economy penalties, use with engines of optimum efficiency, in-vehicle space, provision of extra air and exhaust gas recirculation, cold start, adaptability to

differing regulations, durability, adaptability to vehicles already in use, and to existing vehicle manufacturing techniques. Because so many of the long outstanding problems of the industry can be overcome by the disclosed means, it is considered that eventually most practical emission control devices will be along the lines of the present invention.

**PORT AREA CONFIGURATION**

This section deals mainly with exhaust port embodiments. How the invention affects inlet port considerations is described subsequently. It is intended that the features described herein may be used in any convenient combinations.

As has been noted, the basic embodiment involves the placing of an open-sided chamber against the engine or cylinder block, so eliminating the conventional exhaust manifold. In effect, the block therewith forms part of the reactor housing, and as such may play as important a role in the reduction of pollutants as the portions of the reactor assembly so far described, namely the applied housing and the filamentary material. It has been shown how the housing fits directly onto the block, whether or not this has other features, such as port liners or filamentary spirals. In alternative embodiments, an intermember may be applied between block and reactor housing proper, this intermember either wholly or partly completing the definition of reactor volume. Where a section ceases to be an intermember and becomes an appendage to the block is not strictly definable, but in general an intermember is considered making contact with the periphery of the housing. The various features described, whether in relation to intermembers or attachments to the block, are intended to be applicable to both, and also where suitable to the periphery of the housing.

The arrangement of the reactor assembly in the manner described affects an art not strictly the subject of the present invention, namely that of exhaust gas flow.

This art has for long been associated almost exclusively with the movement of columns or pistons of gas, and in particular with the kinetic energy and pulsing effects which are built up in the regular dimensioned columns of gas. The present invention dispenses entirely with regular tubular configurations in the exhaust system's initial and most important section, with the result that the exhaust gases will flow in a manner previously little explored. Initial research has indicated that the gas flows of the invention present possible benefits. Firstly, the relatively great increase in cross-sectional area of the reaction volume over the total cross-sectional area of the exhaust ports ensures a considerable decrease in the velocity of the gases. The reduced velocity will greatly lengthen the durability factor of at least parts of the reactor assembly, since much wear is caused by the abrasive effect of the fast moving gases and their particulate content. Secondly, the gases from each cylinder meet and merge in the reactor volume, eliminating exhaust pipe branching. Branching is one of the problem areas of conventional exhaust flow art, since it is here that considerable power losses often occur. It is possible by careful design of branches to eliminate much power loss, but usually only within an optimum flow range. When engine speed varies above or below this, power losses increase. Thirdly, the reaction volume will, to a valuable degree, absorb vibration and, as has been mentioned earlier, also sound. Conventional exhaust pipes, with their regular, tubular configuration and metallic construction, may transmit and be the cause of, usually thorough magnification, of much vibration in their own right. The vibrations originating with engine combustion and carried by the exhaust gases will tend to become dissipated by the large volume of gas and filamentary material in the reactor. Although it is useful to place the reactor over a conventional exhaust port exit with its cylindrical shape it is felt that the sudden transformation of the gas from a columnar configuration to the amorphous flows of the reactor volume, plus the sharp edge of the junction between port and block will together contribute to an unnecessarily inefficient gas flow and consequent power loss. For this reason, in a preferred embodiment the exhaust port bells out, that is progressively increases its diameter in

some manner, and has been so shown in the sections of Figures 3 and 5. This has the beneficial effect of decelerating the rate of gas flow progressively.

In Figure 6 is shown diagrammatically a housing 51 enclosing a reaction volume 52, both having interposed between them and engine block 53 with exhaust port 54 an intermember 55 of substantially flat configuration. Figure 7 shows a similar arrangement, but with the intermember 55 in association on one side with both engine block 53 and an exhaust port liner 56, which in the embodiment illustrated is restrained in position by the intermember 55. Figure 8 shows a similar arrangement to that of Figure 6 but with the substantially flat intermember 55 recessed into a corresponding depression 59 in the engine block 53, being restrained against the block in the embodiment shown by the enclosed housing 51. In Figure 9 is shown an arrangement similar to that of Figure 6 but where the intermember 58 is of enclosing configuration, that is when viewed in elevation from the reaction volume side it is seen to have a depression 59 defined by a peripheral lip 60, the outline of which corresponds with that of the lip 61 of the enclosed housing 51. A notional plane drawn across the lips will define two sections of the working volume of the reactor, one within the housing at 62, the other within the depression 59, of the intermember. Figure 10 shows a broadly similar arrangement, but where the mounting between housing and intermember is used to support filamentary material 63. Figure 11 shows an arrangement similar to that of Figure 9 but where the enclosing intermember 64 has an integral projection 65 on its engine side, in this embodiment of approximately ring or hollow cone like configuration, to act as exhaust port lining. Figure 12 illustrates the fixing detail at (A) in Figure 6, where an L clamp 66 and bolt 67 press the housing 51 to interplate 55 and thence to engine block 53. Compressible heat resistant material 68 is interposed between the joints to allow for proper sealing, possible differential expansion of the various components and more even load distribution between possibly marginally mismatched surfaces. Figure 13 is a detail at (B) of Figure 8 showing a similar fixing technique, and an alternative embodiment where the interplate 55 retains in position an

exhaust port liner 56. Figure 14 shows a fixing detail similar to that at (C) in Figure 10 but retaining a different type of intermember 69, one which does not substantially mask the engine block, but which is part of an effective division of the enclosing housing, the advantages of which are explained below. Here the two portions are shown separately fixed to the block, although in some embodiments only the outer housing need be fixed, depending on detail design. By example, the housing 51 is retained against the intermember 69, by means of strapping band 70 pivotally attached to winged extensions 71 of a collar 72 mounted on unthreaded portion 73 of a stepped diameter stud 74, by means of nut 75 and washer 76 shown dotted. The intermember 69 is fixed to the block 53 by means of the same stud 74, an L clamp 66 and a washer 77 and nut 78 of larger internal diameter than the set 75, 76. Compressible heat resistant sealing material 68 is disposed within the joints between mating surfaces.

The provision of an intermember may have at least three principal advantages. Most importantly, it offers an opportunity to prevent heat loss from the reaction volume to the metal engine block and its associated cooling system, since the intermember can be made of insulating materials such as ceramic, similar to those of the main housing. Secondly, the additional and more conveniently disposed joints between various pieces may be used also to act as supports for additional matter, such as the filamentary material 63 between intermember and housing in Figure 10 and between intermember 55 and block 53 in Figure 7. Thirdly, the intermember offers the opportunity of splitting a reaction volume housing whose internal (or external) surface describes a curve of more than 180 degrees in cross-section, so that the portions may be manufactured on a male (or female) mould, a possibly cheap and structurally desirable way of producing the housings. It can be seen, for example that the reactor of Figure 10 might not be manufactured by molding if it were of integral construction in cross-section. Although in each case only one intermember has been illustrated, a plurality of intermembers may be used in association with one enclosing housing, or multiple intermembers may be combined to form such a housing.

Figures 15 and 16 show diagrammatically by way of examples sectional plan views of reactor housings 79 mounted over the exhaust ports 54 of an engine block 53, where depressions 80 have been formed in volume usually occupied by the engine block assembly, the space gained by the depression becoming an integral part of the reaction volume 52. In Figure 15 there is a continuous depression and in Figure 16 a series of depressions have been formed about provisions for twinned inlet ports at 81. Apart from the two above examples, space normally occupied by engine may be given over to the reaction volume in any configuration. It is generally desirable to have reaction volumes as large as possible for purposes of exhaust emission treatment, the limiting factors often being lack of under-hood space in vehicles and the cost of providing greater and stronger reactor housings. In the case of the present invention, reaction volumes may be increased without any sacrifice of under-hood space or increase of housing size and cost, by the procedure of "hollowing" into the engine block. The degree to which this will be possible will depend on such factors as whether an engine is especially designed to accommodate the invention or not. If so, it will have been possible to greatly reduce the water-jackets (if water cooled) in that area, especially if insulating exhaust liners are incorporated, since it is desirable in the case of the invention to as far as practicable eliminate heat loss in the exhaust area and liners will obviate the need for cooling. Hollowing into the engine block is a means to allow more progressively shaped reaction volumes and more efficient and smooth gas flows to be achieved.

Figure 17 shows by way of example a diagrammatic sectional plan view of a reactor housing 79 mounted on an engine block 53, having exhaust ports 54 whose axes 82 are not parallel to one another and/or not perpendicular to the block face, while Figure 18 shows a similar arrangement in vertical cross-section. It is important that the exhaust gases distribute themselves as evenly as possible within the chamber so that the factor of time, multiplied by the area of surface exposed is as equal as possible for the gases from differing ports, and that such wear and/or loading caused by

abrasion, corrosion and gas velocity is as evenly distributed within the reactor as possible. This optimum equalling out effect may be achieved, among other means, by angling the flow from each port in the most suitable directions, which will often involve port axis layouts along the lines of the example described by Figures 17 and 18. In this preferred embodiment the end port axes are furthest angled from the perpendicular to engine axis in plan view and the central port axes furthest from the perpendicular in vertical cross-sectional view, which will enable the gases to more readily travel the same distance to the reactor gas exit. Below is mentioned an alternative or complimentary means of better distributing gas flow.

It has been seen in the basic embodiment, described in the previous section, that filamentary material may be introduced in the exhaust port area, both to assist in the process of reaction and/or to properly direct the flow of exhaust gases. The control of gas flow may be achieved by providing members of substantially vaned, honeycombed or flanged configuration within the port, such members being manufactured of any suitable material such as metal or ceramic, but according to current technology are preferably made of metals having catalytic effect such as nickel/chrome alloy if the gas flow directors are desired to significantly assist in the reaction process. The particular embodiments of filamentary material suitable for exhaust port areas, with their relatively restricted cross-sectional areas and high gas flow rates (compared to those of the reaction chamber itself), are those where the material does not have significantly great cross-sectional area, which would cause obstruction to and acceleration of gas flow past the material. However, any configuration of filamentary material may be employed in the port area, including the various embodiments described subsequently, especially if it is intended to utilize the material to assist in the reaction process.

By way of example, there is shown in Figure 19 in cross-sectional view and in Figure 20 in front elevational view as seen from E, an exhaust port liner

combined with honeycomb configuration gas flow director 83, 85 and held in position against engine block 53 by intermember 55, there being heat resistant compressible material 68 between the joints. Inside the port 54, the greater mass of gas will be concentrated toward the outside of the curve at 84, and therefore the honeycomb structure has at the end facing the gases a diagonal face across the port as shown, so that whatever frontal area the honeycomb vanes 85 have will cause the gases by deflection to pass through the structure more evenly distributed. With progression of gas flow the vanes become more mutually further spaced, so reducing gas velocity, and curve away from each other, so that the mouths 86 of the structure will direct the gases in a multiplicity of directions. The honeycomb structure may be of any suitable cross-sectional configuration, including by way of example, that of Figure 21, where the passages have six faces, or that of Figure 22, where the passages are formed by the intersection of radial and coaxial membranes. In an alternative embodiment, gas flow is directed by flanged members running part of the length of the exhaust port, as shown by way of example, in an embodiment illustrated in sectional plan view Figure 23 and in partial cross-section in Figure 24. The flanged members are alternatively "Y" shaped configuration at 87 and of roughly cruciform configuration at 88 and are spaced and held from each other by spacer risings 89 disposed at intervals along the length of the assembly. The flanged assembly of the illustrated embodiment is retained by fitment into grooves 90 in the port surround 91, such grooves optionally containing a compressible bed 92 at F in Figure 23 and are held against block 53 by intermember 55 sandwiching the bent extension of flanges as at 93 through compressible material 68.

It may be desired to impart a rotating motion or swirl to the exhaust gases during their passage through the ports, so as to assist in the proper mixing of gases within the reactor volume. To this end, successive ports may have alternating directions of swirl, as indicated diagrammatically in Figure 25. The swirl may be imparted by vaned members disposed diagonally across the axis of gas flow. The vanes may be placed anywhere within the port area but in a preferred embodiment illustrated

diagrammatically in Figure 26, the vanes 94 project from and are integral with the exhaust port wall or lining 95. If it is desired to introduce some turbulence as well as swirl to the gases, the individual vanes may be of waving configuration, as shown by way of example elevationally in Figure 27 and in Figure 28 in a sectional plan view through G of Figure 27.

All the features described herein may be combined in any convenient or desired way. By way of example, Figure 29 shows a preferred embodiment in cross-section. The reaction volume is enclosed by an intermember 55 of ceramic material having projections comprising exhaust port liners 56 and spaced from engine block by compressible heat resistant material 68 such as ceramic wool, together with an enclosing housing 51 of integral ceramic construction. The joint between the two principal enclosing members supports a filamentary space frame 96 that is a construction of short straight metal rods connected to each other at different angles, which substantially fills the foremost part of the reaction volume, the rearmost portion of which is occupied by filamentary material of wool-like configuration of say a ceramic based compound. Within the exhaust port area are two metal cone shaped spirals 97 the free ends at their cemented back to back meeting projecting to from bayonet fixings shown dotted at 98, which locate in grooves 99 running from initial entry away from the direction of the exhaust valve, so that the pressure of gas flow will cause the spring projections or bayonets to seat at the end of the groves.

Throughout this disclose the expression "engine block" is meant to denote what is known as either an engine block or a cylinder head block in conventional motor usage.

FILAMENTARY MATERIAL

This section deals almost exclusively with the alternative forms of filamentary matter, its material composition being described subsequently. Filamentary material was defined as portions of interconnected material which allow the passage of gases therethrough and induce turbulence and mixing by changing the directions of travel of portions of gas relative to each other. By interconnected is meant not only integral or continuous, but also intermeshing or interfitting while not necessarily touching. The above definition is applied to material within the reactor as a whole, not necessarily to the individual portions of that material. It is especially envisaged that in its most effective form the filamentary material in one reactor will consist of sections of varying composition. The three main classes of filamentary material may be said to comprise slab or sheet material, wire and wool, and are listed in order of progressively less resistance to abrasion and shock, provided of the same material. Therefore it is logical to place the more robust forms nearer the exhaust ports with the more fragile embodiments toward the rear of the reactor. If catalytic effect is desired, then the most suitable materials may be best incorporated in a particular form, this form being such that it is most suited to be placed in a particular portion of the reactor. It is possible that more than one catalyst is desired and these may be incorporated in positions most suitable to their differing forms. It has been noted that the main chemical reactions tend to take place in a certain sequence and, if special catalytic assistance is desired for a particular reaction, that catalyst in combination with the most suited form of filamentary material may be placed in that area of the chamber where the reaction is most likely to occur. For example, if the reaction in question is expected to be the last to take place, then the appropriate catalyst/filamentary matter will be disposed in the rear half of the reactor, furthest from the exhaust ports. The definition of filamentary material, therefore, is meant to apply to that within the reactor as a whole. and not necessarily to each of the possibly many varied components that may make up one reactor filamentary assembly.

It is intended that the various embodiments of filamentary material described may be combined in any convenient manner within a single reactor assembly.

By way of example, an embodiment is shown cross-sectionally in Figure 30 and in part sectional plan view in Figure 31, wherein alternate slabs of honeycomb structure 101 and wool-like layers 102 make up at least the rear portion of a reactor 100. The path of a certain pocket of gas through the system is indicated in each view by the arrows 103. It will be noted that the honeycomb is not of conventional form, since it consists of passages with each stack or row of passages running in a different direction from the adjacent row. In the first honeycomb slab 104 the passages shown in section 106 run "downwards" while the passage immediately behind, shown dotted at 107, are running "upwards," with the separation of direction and therefore of gas flow taking place substantially in the vertical plane. The next honeycomb slab, 105 is of the same construction but placed turned through ninety degrees, so that the separation of gas flow is substantially in the horizontal plane. In this way the different portions of gas are properly intermixed, as can be shown by the path 103a shown by dotted arrows of a gas pocket starting adjacent to the first pocket and in its path through the assembly becoming widely separated from it. In other words, although an individual honeycomb passage does not induce turbulence, the disposition of passages relative to each other does so within one honeycomb structure, as may the provision of a succession of honeycomb configurations placed one behind the other.

A form of filamentary material, not strictly wire or slab, which may be successfully employed in the invention is expanded metal or metal mesh. By way of example Figure 32 shows in diagrammatic sectional plan view how sheets of metal mesh formed into wavelike configuration are placed one behind another inside a reactor 100, while Figure 33 is a detail enlargement at H showing construction of the mesh. Mesh is usually formed by a combination of pressing and tearing sheet, processes which tend to leave sharp edges. Because materials are less resistant to heat,

abrasion and corrosion when they are not smooth and rounded, the mesh used should preferably be subjected to sandblasting or other smoothing process after forming. Metal mesh is a known product and could readily be manufactured of catalytically active metals. The particular forms described may also, because of their inherent suitability to the invention, be manufactured of non-metallic materials such as ceramic, possibly by alternative forming means.

Filamentary material in wool-like or fibrous configuration is especially advantageous in the invention because of its ratio of high surface area to mass and because it will more readily act as a particulate trap. Catalytic agents may be deposited on surfaces, for example by precipitation or deposition processes including those involving immersion in solutions or other fluids. If the material itself is to have catalytic effect, it will most readily be manufactured of metal, to which the considerations above will apply. That is, it should in the interest of durability be as smooth and rounded as possible, the wool preferably consisting of multiple fine regulation wire, woven, knitted, layered or randomly disposed. If the wool is composed of say fibers or strands of such materials as ceramic glass, this will be more temperature, abrasion and corrosion resistant than metals, but will be more susceptible to "flaking," that is particles or whiskers becoming detached from the general mass by the force of the gas flow, to perhaps lodge in a sensitive area downstream such as a valve. For this reason it is preferred that wools are placed in the sections of the reactor most suitable to them, in the case of metals rearward away from the full heat and force of the gases, and in the case of ceramic fibers distanced from the gas exit. Alternatively and preferably, wools should be sandwiched or contained by other forms of filamentary material, for example as in Figure 30.

Another appropriate form of filamentary material is wire, especially since in the case of metals it is almost always readily available in that form and need only be bent or otherwise formed to any desired shape. For reasons of durability, the wire

deployed generally needs to be thicker nearer the exhaust gas source than elsewhere in the reactor. The wire may be woven 108 or knitted 109 into a mesh as illustrated diagrammatically in elevational section in Figure 34. It is preferable to devise a deployment of wire which avoids normal contact between strands, because the vibration of the internal combustion engine will tend to cause attrition at the point of connection, perhaps resulting in premature failure. Therefore the wire should preferably be arranged in forms to enable a relatively great length (i.e., surface area which is assisting reaction) to be incorporated in the overall restricted area of the housing with the various portions of wire having minimum contact. It is expected that some contact will take place between wires spaced close together but not touching, but this contact should preferably not be regular, although its occurrence during freak vibration period or operating modes should not materially affect durability. An obviously suitable way of deploying the wire is in the form of spirals or coils, shown diagrammatically in elevation disposed axially across the flow of gas in Figure 35, and disposed coaxially with the flow of gas in Figure 36. By way of example spirals having regular coils of equal diameter are shown at 110, while those having regular coils of progressively varying diameter are shown at 111, and spirals having irregular coils, that is of non-circular configuration and/or random diameter at 112. The three configurations comprise spirals having axes of substantially straight line configuration. Figure 37 shows in diagrammatic cross-section spirals 113 having curved axes, here arched to better withstand force of gas flow from direction 114. Any of the spiral types mentioned previously may have curved axes. The wire may also be disposed in two or three dimensional snake-like configuration. Such a two dimensional form is shown by way of example diagrammatically in elevation in Figure 38, while a three dimensional form is similarly shown in elevation in Figure 39 and plan view in Figure 40. Such forms may be disposed within a reactor in any number of ways, as for example shown in diagrammatic sectional plan view in Figure 41, where flat "snakes" 115 and curved "snakes" 116 (each snake comprising wire looped in the plane indicated) are stacked next to each other and behind each other, either spaced as at 117 or intermeshing as

at 118. These stacks of loops or curves may also be randomly placed (not illustrated). Figure 42 shows diagrammatically how the plane of curves 119 may be straight, or as in Figure 43, curved as at 120, to withstand gas flow from 114, or as in Figure 44 curved as at 121 to provide a more ready and natural path for the gas flow. Figure 45 shows in similar view how the planes of snake-like loops or curves, whether curved as shown or straight, may themselves intermesh past each other in any one or more dimensions, where the planes in solid line 122 are in the foreground and planes shown in dotted line 123 in the background. Figure 46 shows in diagrammatic sectional elevation how the planes of curves, as viewed head on, may intermesh in other ways, where 124 are planes shown solid in end elevation (here curved in a third dimension, although they may be straight) slanting across the path of planes behind shown dotted 125 running in other directions. Alternatively, their curvature in the third dimension may be non-coincidental, as shown at 126, while at 127 is shown how the curves in the third dimension allow for the close stacking of these planes. Conveniently, the planes span the shorter dimensions as shown, but they may also span the longer dimension. Alternatively, the wire may simply be disposed in strands across the reactor, as shown by way of example in diagrammatic elevation in Figure 47, where wires in the foreground are shown solid 128 and those behind dotted at 129. To assist in the elimination of sympathetic vibration, the various strands may be not quite parallel that is at a slight angle to one another (not illustrated). Generally, because the strands of the latter configurations may be arranged to be in tension, they need be of thinner configuration than the largely self-supporting structures such as spirals or snake-like loops. Wherever wire is herein described it is meant to comprise either a single strand, or multiple strands, as for example in diagrammatic section Figure 48. Because the material preferably exposes the maximum surface to the flowing gases, it may be desired to separate the individual strands of the wires to allow gas to flow through and past each strand, but to simultaneously still allow the separate strands to a degree support each other. Conventional separators may be employed, e.g., of ceramic, but in another embodiment the individual wire is crimped. that is minutely and closely bent in

all directions, as shown elevationally in Figure 49. As can be seen in cross-section Figure 50 the wire effectively occupies a greater diameter, shown dotted, than its real thickness, resulting in the composite wire of Figure 51. Fixing of wire and other filamentary material to reactor housing will be described later in this section.

The filamentary material may further comprise sheet or slab, and in a simple form may be described as a plane having some thickness, in the same way as did the series of snaked wire loops. These planes may be disposed within the reactor in much the same way as were those of the wire loops as described above. For example, the planes may comprise long sheets, straight or curved and be disposed as illustrated diagrammatically in Figures 41 to 46. Such sheets may further have a form of simple alternate wave as shown in diagrammatic cross-section in Figure 52, or a more complex waved or dimpled form as in Figure 53. Alternatively, the sheet may have a sharply curved or crooked cross-section, as in Figure 54, to present a greater frontal area to gas flow 114. The sheet may further be in the form of holed fins or vanes as in cross-sectional Figure 55, preferably having a thicker, more rounded section toward the side facing the gas flow 114. The holes in the sheet may have pressed projecting lip or lips, as shown in Figures 56 and 57, or the holes may comprise apertures formed by punching, pressing and/or tearing, without significant removal of material, as shown for instance in cross-sectional view in Figures 58 and 59. Figure 60 showing a part elevation of such a sheet, illustrates diagrammatically examples of forms of holes or pressed/torn apertures. Again, preferably sharp edges are removed after forming by blasting or other means. The sheet or slab may be formed into three dimensional interlocking or intermeshing forms, as shown by way of example in sectional elevation Figure 61, where 130 describes a series of interlocking rings and 131 a series of interlocking hexagons. Figure 62 is a diagrammatic cross-section showing by way of example a pattern of interlocking here using conical rings 132. Figure 63 similarly shows interlocking means, but here the overall form is curved rather than linear. Figure 64 shows in diagrammatic cross-section how individual sheets 133

interlock to make up a three dimensional form, while Figure 65 similarly shows the employment to this end of curved sheets 134.

The filamentary material may be fitted to the housing in a number of ways. Considering Figures 66 and 67 both sheet or slab 139 and wire 136 whether part of looped or spiral forms, or as in Figure 36, wires 135 acting as structure supports may lodge in recesses 137 in the housing 138 as in detail section Figure 66, or may be gripped by protrusions 140 as shown in detail section Figure 67 and plan Figure 68. Compressible material 141 may be interposed between filamentary matter and housing to prevent attrition through vibration. Alternatively, sectional plan Figure 69 and elevation Figure 70 shows how sheet 139 may be connected by linking members 142 which in turn affix to housing 138 along the lines illustrated in Figures 66 and 67. However, if the sheet is of suitable material such as ceramic it may be incorporated into the housing during the manufacturing process of the latter. By way of example, sectional plan Figure 71 and elevation Figure 72 show how slab 139 having appropriate, preferably holed, linking members 142 is integrated with housing 138 by means of the shrinking during formation of the housing in still malleable form upon the pre-formed prior-positioned interlinked slab assembly. Such a technique is considered especially viable where both filamentary material and housing are to be formed of ceramic.

The filamentary material may further be of the random or regular forms resulting in certain manufacturing processes involving what can roughly be described as fluid collision, possibly in association with reduction techniques. This will be more fully described subsequently. A further form of filamentary material has already been alluded to in the previous section, and that is the space frame, a construction formed of short, usually straight, mutually interconnected rods.

The filamentary material may further be in the form of pellets, preferably in spherical form, or occupying a theoretically spherical form. Pellets are known in the

art, comprising small regularly surfaced globes. In alternative embodiments the pellets may be of irregular semi-ovaloid form as in Figure 125, or of roughly kidney or bean-like configuration as in Figure 126. However, it is preferred, so that most advantageous ratio of surface area to mass may be obtained, that the pellet comprises a form consisting of a series of projections and depressions, this form most conveniently having an overall spherical aspect, and configured so that preferably the projection of one pellet may not too easily fit into the depression of another pellet. If such interfitment is kept to minimum, it will ensure that the pellets are not tightly against one another, and so ensure a proper easy gas flow about and between the pellets. Figure 127 shows in sectional elevation by way of example such a form, having four equally spaced projections 390 radiating from a central core of roughly mushroom or bulb-like configuration. (Forms similar to this are used in concrete blocks for breakwater construction.) The same principles might be applied to a pellet having a greater number of projections as shown diagrammatically in Figure 128, or having a multiplicity of projecting vanes, preferably disposed at angles to one another to better space adjacent pellets from one another, as shown in Figure 129. In Figure 129a the pellet may consist in a sphere having substantial snake-like depressions of rounded cross-section disposed in its surface. An embodiment similar to that of Figure 127 is shown in Figure 130, where the projections 391 are of more pronounced mushroom-like shape. Such pellet-like material will assume its most possibly compacted form under vibration, rather than when being fitted. To ensure that the pellets remain, after initial settlement, in a basically constant physical relationship to each other (rather than excessively move about and so wear more rapidly) the pellets are best subjected to some continuous pressure. This can for example be achieved by mounting pellets between filamentary material of wool and/or wire configuration. For example in cross-section Figure 131 a housing 392 encloses pellets 393 adjacent to wool 394, in turn adjacent to wire 395.

The filamentary material may further have an ablative effect, that is its decomposition may be desired and controlled, in this case to contribute therewith to the desired reaction process. A material may be used resulting in the filamentary matter having a deliberately limited life span and providing within the reactor a compound which will react with the pollutants and/or gases under certain conditions.

## COLD START AND RELATED FEATURES

It is proposed in this section to deal more fully with the various aspects of the cold start procedure, including the sequences and activating means of the valves. the possible ways the closure period may be prolonged without consideration of interference with engine settings, different forms of valve construction and lastly a brief mention of how valve configuration may have unconventional embodiments including those used for gas recirculation.

It has been seen in section one that, for the cold start operation to be effective the gas exit valve must be closed for as long a period as possible, the so far limiting factor being the amount of pressure attainable in the reactor without stalling the engine. In some case, when the reactor has exceptionally rapid warm up characteristics it will not be difficult to keep the valve closed until the threshold of operating temperature is reached. With other systems it will be more difficult, if not impossible. In such cases, it may not be advantageous to partly open the gas exit thereby maintaining the pressure, since the gases emerging will only be partly de-polluted. As an optional alternative therefore, it is proposed that there be fitted to the reactor a passage communicating with an exhaust gas reservoir, and that there, optionally, be a second independent closure means between reactor and reservoir, preferably near the junction of passage and reactor. In operation, when the acceptable level of pressure in the reactor is reached (including a pressure no greater than atmospheric), the gases

pass through the passage, either because there is no obstruction or because the obstruction to the reservoir has been removed. Once reactor warm up temperature is attained the flow of exhaust gas to reservoir would substantially cease. The gases are then expelled from the reservoir by any means, but preferably during the operation of the car while warm, either to the engine intake system and be recirculated through the combustion process, or to the reactor which being warm would satisfactorily process them. Because the gases are always continually reacting, however slowly, it is likely that they would become significantly pollutant-free during their sojourn in passages and reservoir. The period of this sojourn is likely to be many times greater, perhaps more than a hundredfold, than the duration of gas passage through the reactor during normal operation.

By way of example, Figure 73 shows in diagrammatic sectional elevation, the engine compartment 152 of a motor vehicle 153 fitted with the reactor 151 of the invention to which is coupled an expansible exhaust gas reservoir 150. Figure 74 comprises a frontal sectional elevation, wherein the left half shows the reservoir expanded and filled with exhaust gas and the right half the reservoir reduced and relatively empty. With an over the reactor 151 is incorporated an inlet manifold 154 surmounted by a carburetor 155. A fan 156 draws air through the radiator 157. The reservoir 150 comprises a folding bellows member 158 mounted on a base 159, the bellows having at the end opposite the base (the lower end) an integral T-shaped stiffening member 160 which communicates at each end rigidly by means of triangulation members 161 to a slidable guide 162 mounted on a vertical rail 163. The bottom of each guide communicates with a compression spring 164 in turn communicating with the lower part of the vehicle structure 165. From a junction 167 upstream of the main reactor gas exit valve 166 a passage 168 communicates with the reservoir base 159 and from this base a second passage 169 in turn communicates with inlet manifold 154. The reservoir is in the position shown so that in normal use, that is when retracted and empty, it occupies a relatively protected position.

In operation, after the main valve 167 has closed, exhaust gas will travel down the passage 168 to fill the reservoir 150. A build up of pressure will be caused because the reservoir can only expand against the force of springs 164. The communication between the reservoir and inlet manifold being unobstructed, the gas will escape into the manifold at a rate in proportion to the size of opening and pressure in the reservoir. When the reservoir reaches a point near the limit of its downward expansion (allowance being made for safety margins) the main valve 166 opens, either partly to maintain pressure if full operating temperature has not been reached, or fully. In the embodiment the aperture between passage 169 and inlet manifold is made very small so that, even under the maximum designed pressure of the exhaust reservoir system, the rate of gas flow into the manifold is very low in proportion to flow produced through the exhaust ports, thereby giving a very reduced rate of exhaust gas recirculation. After the reservoir has been filled and gases diverted down the normal exhaust system, the loading of the springs 164 will ensure the slow collapse of the bellows 158 and the continuing bleeding of gas into the inlet system until the reservoir has been emptied. During the warm up period for the engine - longer than that for the reactor - the reservoir is acting as a radiator blind, reducing the degree of radiator screening with progressive collapse of reservoir, which matches the increasing warm-up of engine system. In usages where wide ambient temperature ranges are found, the size of aperture to inlet may be selected by engine operator, so that for instance in cold winter conditions, the degree of gas recirculation may be reduced by selection of smaller opening to give effectively a larger radiator screening period. The provision of a second valve communicating with passage 168 may in some configurations be omitted by the provision of a relatively small opening between reactor and passage at junction 167, the opening being of many times smaller cross-sectional area than the main exhaust pipe 170. The smallness of opening will restrict gas flow from reactor during the initial stages of warm-up and main valve 166 closure, until the higher pressure in the reactor accelerates the rate of gas flow along passage 168 to more rapidly fill up the reservoir. The non-closure of the small opening at 167 will ensure that the exhaust

gases will effectively be recirculated to the reactor once normal warm operation commences. Depending on the strength of reservoir springs 164, the gas flow rates back through the opening will be lower than those into the reservoir, since the pumping action of the engine must necessarily have considerable greater force than spring action. If it is considered that the gases diverted to the reservoir system have not sufficiently reacted by the time they re-enter the reactor, then catalytic material may be associated with the reservoir, or its internally faced components and/or those of passage 168, 169, or they may be fabricated of a material having catalytic action, such as copper or nickel. Alternatively or additionally, junction 167 may be placed as closely as possible to the exhaust ports, so that the returning gases travel through a substantial portion of the now warm and fully operative reactor. The reservoir assembly may be made of any suitable materials, which to a degree will need to be heat tolerant. If the chosen materials have low heat tolerance, then optional heat dispersal means may be affixed to passage or pipe 168, as shown diagrammatically at 171. If materials are heat resistant, as for example would be a bellows assembly made in silicone rubber, then insulating means may be incorporated on the passages, as shown diagrammatically at 172, with the advantage that the gases may be maintained in the reservoir at warmer temperatures, thereby speeding up reaction processes. The warmth of the gases may be used to advantage in another configuration, where the gases are recirculated to the intake system above or at the carburetor. The provisions of this flow of warm gas during cold start - as has been shown above, the reactor may be operative to a degree already from a few cycles after firing commences - will assist in vaporization of fuel during engine warm up. In a preferred embodiment, the gas is recirculated directly through the choke or cold start system of the carburetor. In normal usage, the gases will not at inlet entry point be hot enough to present risk of premature fuel combustion. Optionally a valve may be provided between reservoir and inlet system to regulate circulation.

The valve construction presents possible problems, since it needs to be tolerant of the very high temperatures and abrasive qualities of exhaust gas, preferably for the full life of the engine. A range of suitable high temperature materials, including ceramics or nickel alloys, are described in more detail subsequently. It is here proposed to describe, by way of example, certain methods of valve construction which entail easy service in the event of need for replacement or maintenance, and which are capable of providing proper sealing, optional diversion of gases to storage or recirculation and some tolerance of particles or whiskers from any filamentary material. The principal feature of the major embodiments herein described is that the joint or flange between two principal components coincides with the valve axis, enabling valve and spindle to be manufactured as an integral unit and fitted when the two components are mated up, this configuration being particularly suited to butterfly valves. Alternative butterfly valve constructions involve the fitting of valve through say a slot in the spindle and location by screws when it has been positioned by passing through integral apertures, as for example in carburetor construction, and it is felt that such alternatives are too delicate or susceptible to expansion problems to be ideal for exhaust gas usage. The coincidence of joint with valve center also results in possible sealing benefits, as will be seen later.

Figure 75 shows by way of example in diagrammatic plan view a reactor component 180 having at its junction with exhaust pipe 181 the main gas exit valve 182. while Figure 76 similarly shows a reactor component 181 having between exhaust pipe 181 and main valve 182 an intermediate section 183 having at its junction with passage 184 communicating with recirculation system an option secondary valve 185. Figures 77 to 81 show details of the valve 182 of Figure 75, where Figure 77 is a sectional view along K, Figure 78 an enlarged plan view, Figure 79 an elevation at L, Figures 80 and 81 details at the joint between sections. Manufactured integrally with spindle 186 and actuating lever 187a is a butterfly diaphragm 187 of biased oval configuration having one section 188 of greater surface area than the other 189, so that the valve will tend

to fail-safe in the open position. The cross section of the exhaust pipe 181 and reactor component near the joint is substantially of similar oval configuration to valve. Both major sections have their jointing and integral flanges 190, which are linked with coincident hollow load distributor ridges 191 through which pass the bolts 192, washers 193 and nuts 194 holding the two components together under compression, separated by compressible material 195 preferably in two separate layers passing each side of the spindle 186. This is shown in detail cross-section Figure 81 through spindle at its passage between the two major components 180 and 181. Preferably the components and spindle should as shown have mating curves of non-coincident centers when assembled, so as to provide a stronger pinching effect in the area of joint 196 where the seal can be expected to be weakest. The slight internal projection of the twin layered compressible material 195, as shown in part section Figure 80, will assist in the proper location and sealing effect of the diaphragm 187 when in the closed position.

Figure 82 shows by way of example a diagrammatic sectional plan of the arrangement of Figure 76, where the optional secondary valve is in the form of a pressure sensitive plug 197 and compression spring 198 assembly, and where a honeycomb structure 199 is located by the junction of intermediate section 183 to reactor 180, in order to act substantially as a fiber or strand trap. Figure 83 shows a similar detail elevational plan view, wherein the passage 184 is joined to intermediate member 183 by at least two assemblies comprising two coincident hollow load-distributor ridges 191 and bolts 192, washers 193 and nuts 194, while the exhaust pipe 181 is connected to reactor 180 through the intermediate section 183 by means of assemblies 200 comprising three coincident load distributor ridges and associated fasteners. Figure 84 shows diagrammatically in longitudinal cross-section a hollow ball valve in the open position fitted in the joint between two components, where 201 comprises the "ball" with its integral spindle 202 and actuating lever 203, with 204 the main exhaust passage, 205 the seals, 206 an optional secondary passage allowing exhaust recirculation means during cold start, 180 the reactor housing and 181 the exhaust pipe, with the joint

between the two shown dotted at 207. Figure 85 shows in similar sectional plan view the above arrangements with the valve in the closed position, allowing the secondary passage 206 to communicate into the main passage 204, which in turn communicates with an aperture 208 leading to exhaust gas recirculation means.

It is desirable to make the valve actuating means as simple and as fail-safe as possible. To this end, the valve should be spring loaded (not locked by mechanical action) in the closed position in such a way that reactor pressure over the designed limit will overcome the force of the spring sufficiently to allow some gas to escape, thereby again lowering pressure to below that required to actuate the spring and maintaining a balance of loading to keep the valve slightly open, to sustain constant pressure in the reactor. The spring loading is such to also bias the valve to the fully open position. Such an arrangement is illustrated by example diagrammatically in Figure 86, where 210 shows a valve actuating lever in heavy line, butterfly valve 211 and internal face of passage 212 in light line, spring 213, spring axis 214 and spring anchorage 215 on housing and anchorage 216 on lever, with pivotal valve axis at 217. The valve assembly is shown in slightly open position on dotted line and fully open in chain dotted line. The same system of loadings may be employed and the valve actuated by making the previously fixed spring anchorage point 215 movable as in the path indicated by dashed line 218 between extremities 219 and 220, dashed line 214 indicating spring axes at each extremity. This movement of spring anchorage may be actuated in any way, and in a preferred embodiment is moved by a member driven by the expansion of heat sensitive material, such as a trapped pocket of gas or was as shown in Figure 87, where a piston 221 communicates with a container of high conductivity 222 exposed to the passage of hot exhaust gas 223 through a volume 224 of trapped readily expansible material such as gas or wax. The piston 221 is connected to rod 225 and linkage 226. Figure 88 shows schematically how the piston rod 225 actuates the operation of the valve by means of its actuating lever 210, spring 213, and an intermediate lever 227 mounted on pivot 228. The actuation of the valve indirectly, by

means of a spring, ensures that fail-safe characteristics are embodied. If this is not considered necessary, then the heat actuated piston 221 may by direct linkage open and close the valve, as for instance if the end 229 of the intermediate lever 227 were connected directly to the valve actuating arm (embodiment not illustrated). In both cases, but especially in the latter, it will be possible closely to relate valve opening to exhaust temperature, and therefore reactor pressure in relation to temperature.

It has been shown that the warm up of the assembly has been hastened by the whole or partial closing of the exhaust gas exit by valves, in effect damming the gases inside the reactor. Such damming may be achieved by any suitable means including, in a preferred embodiment, the provision of a fan or turbine in the exhaust system adjacent to the reactor gas exit. Because the fan is inert on cold start and constitutes a barrier or dam in the system, pressure would build up behind it during the early cycles of engine operation. The fan preferably would not constitute a total barrier, some air passing either between the blades or their junction with housing, enabling the engine to be turned over on the starter motor with relative ease. Once firing commences, the rapid increase in engine speed and gas flow would ensure a considerable damming effect, which would only be relieved when the reactor pressure against fan blades overcomes the fan's inertia. Optionally the fan spindle and its bearing may have differential coefficients of expansion, so that when cold a tighter bearing fit would ensure greater resistance to rotation than when warm.

The above features may be used in any suitable combination with each other and also, where appropriate to fulfill functions not related to cold start. Gas circulation to inlet system may be associated with a gas reservoir, or alternatively it may be direct, that is eliminating the reservoir. Further, the exhaust gas recirculation (ERG) system described previously could for example be used after warm up had been achieved to provide EGR to the engine under normal running, either continuously or under certain operating modes. To facilitate the use of EGR and so thereby possibly

to eliminate the use of pumps, a scoop may be placed in the reactor about the junction with recirculation passage, as illustrated diagrammatically in Figure 89, where the scoop 230 projects into the exhaust gas flow 231 so creating a higher pressure area at 232, which asserts the flow of gas along the EGR system 233. Preferably, the scoop is placed in a "weak" area of the reactor. that is where the reactions are taking place at below average rates, so that the least pollutant free gases are recirculated, permitting the reactions partly to continue during their second passage through the reactor. The scoop arrangement would entail that EGR employed continuously is in roughly constant proportion, after a build up of proportion between very low and medium speeds, since gas circulated depends on speed and therefore volume of gas issuing from the engine. Generally EGR absorbs engine power, but at certain lower rates and/or operating conditions EGR may marginally increase engine power. For this reason, and/or better to eliminate pollutants, it may be desired to have EGR operative under only specific running conditions, such as acceleration or deceleration, etc. An optional valve at junction of EGR system to intake manifold would, by way of example shown in diagrammatic section Figure 90, be intake vacuum operative, where 234 is the exhaust supply passage, 233 the EGR system, 235 the manifold, 236 a plug shown in open position against pressure provided by curved leaf spring 237, but which when closed seals passage 238 provided with progressively sized vent 239 operative when plug is wholly or partly in open position. The plug cap when closed seals against seats 240, where internal volume at 241 is pressure balanced with EGR system by weep passage 242. The degree of EGR in proportion to inlet vacuum (caused by say acceleration if placed before the carburetor valve or by say deceleration if placed between carburetor and inlet port) will be regulated by the sizing of vent 239, which may be of linear, logarithmic or other progressively increasing dimension. The adoption of an operating mode may involve the need for a sudden supply of recirculated gas. With a direct system, once the initial demand has been met, a partial vacuum will be created in the EGR system, thereby slowing down rate of gas supply to below that ideally required. This may largely be obviated by incorporating an exhaust gas reservoir into the system,

which may or may not be expansible. If an expansible reservoir, such as may be used in the cold start procedure is incorporated, then its expansible action may be progressively spring loaded. During normal running, recirculation pressures, say assisted by damming, are in the low range causing the first soft section of the springing to allow the reservoir to expand and contract within a range of say one quarter of its full expansion, this reservoir movement ensuring more consistent EGR rates at the sudden introduction of certain operating modes. During cold start the greater pressures will overcome the resistance of the second stronger section of the springing (as well as the first stage) allowing the reservoir to expand to its maximum capacity.

It has been said that EGR may under certain conditions contribute to marginal increases in power. In fact it is almost impossible for this to be achieved directly; any power gains are caused by the reduction of octane number requirement that EGR results in, thereby permitting increased compression ratios and more optimum valve and ignition timing for a given fuel. Because EGR assists in the prevention of pre-detonation or "knock," it is usually required especially at high load conditions. Previous systems have been proportioned to inlet vacuum, which is not necessarily very great under all high load situations. At least a portion of the EGR system, preferably under low pressure perhaps maintained by a reservoir, may therefore be connected directly to an enriching circuit in a carburetor only operative under high load conditions. Alternatively, an inlet gas velocity actuated valve, as shown in section plan Figure 91 and elevation Figure 92, may be incorporated at the junction of EGR system to inlet manifold. The valve, shown open in Figure 91, comprises a shaft 243 slidable in a passage 244 communicating with EGR system, exposing a progressively sized vent 245, said shaft terminating in a head 246 having attached to it scoops or vanes 247 projecting into the gas stream 248 against the action of looped leaf spring 249. Figure 92 shows the same arrangements with the valve which is accommodated in a housing 250 projecting clear of inlet manifold wall 251 in the closed position. Preferably a properly balanced EGR system will comprise a series of valves, say

actuated by vacuum and/or velocity or other means, disposed in different parts of the inlet system and all communicating with the EGR system, preferably having a gas reservoir. By careful positioning of these valves, regulation of their spring bias to closed and selection of passage diameter, the right amount of EGR could be provided for the various driving modes. In or without association with such a valve system, part of EGR may be passed through certain fuel evaporation circuits, including those described later in this disclosure. The above system of valving and supply, described in connection with the supply of EGR, may also be employed to provide extra air to the inlet system, so as to assist in the provision of a precisely controlled air/fuel mixture ratio, especially desirable in the case of tri-component exhaust emission system. The air may be supplied from a reservoir which has been fed through the air cleaner, as shown diagrammatically in Figure 93, where a coaxial chamber 252 surrounds the main inlet pipe and is adjacent the air cleaner 253, it being supplied with air through opening 254 having optional dams or scoop 255 to maintain air in the reservoir under low pressure. The same system of valves actuated by engine modes could be used to supply recirculated exhaust gas or air to the reactor by means of a passage leading from source to reactor via valve positioned say in air inlet system. The operation of such a valve is shown schematically in Figure 94, where a shaft 256 and head 257 in the inlet system 258 open against spring 259 loading to free passage 260. It is preferred that there is incorporated in any EGR system a filter to trap particulate matter in the exhaust, this matter having been known to lead to increased engine wear and likelihood of mechanical failure in many previous improperly filtered systems. It is felt that with the invention, substantial air supply to the reactor will not be necessary. However, it may be desirable to supply small quantities of air, preferably by means described above, only under certain running conditions to assist in the accurate balancing out of any tri-component process. The air reservoir may be expansible, say by the provision of elastomeric sides, to provide air under more constant pressure with sudden change of operating mode. Alternatively the reservoir may consist of a series of slidably-mounted housings capable of collapsing into one another, for example as

shown in diagrammatic perspective in Figure 144, wherein 600 is the base housing having sides and bottom, 601 an intermediate housing having sides only, 602 top housing having sides and top, with 603 pressed projections acting as guides. The spring loading arrangements and guides disclosed previously may be associated with this reservoir.

## REACTION PROCESS

The health and environmental effects of substances emitted by industrial and automotive exhausts have been extensively researched during the last fifteen years and a great deal of literature exists on the subject. It is known that the primary pollutants of internal combustion engine exhaust gas are carbon monoxide, hydrocarbons, oxides of nitrogen and particulates. These substances are nearly all harmful or undesirable in their own right (carbon monoxide being extremely poisonous) and also combine in the atmosphere in greatly complex reactions with themselves or other substances to form further undesirable substances, part of this process being loosely described under the heading formation of photochemical smog, since the secondary reactions (in the atmosphere) are often activated by sunlight. An outline of the formation of smog and its relationship to pollutants is described for example by Prof. James N. Pitts, Jr. and Gerald E. Grimstone in a paper to the ISAP 1972 Conference in Tokyo. Some idea of the scale of the problem may be judged by the figures for the tons of pollutants per day emitted in the Los Angeles area alone (with percentage attributable to automotive sources in brackets). Hydrocarbons 2,465 tons [65%]; Carbon Monoxide 9.105 tons [98%]; Oxides of Nitrogen 1,050 tons [72%]; particulates 130 tons [42%]. It is felt the subject need not be discussed here because of the comprehensive work of others and its only peripheral relevance to the invention.

The basic principles of the reaction processes of the three main gaseous pollutants when using the tri-component method of approach, plus the basic principles of particulate treatment, have been described under section three, with some additional disclosure in sections five and six. It has been noted that the invention is intended to operate with any desired catalyst and to be adaptable to process the exhaust gas of any internal combustion engine. Where applicable, the principles of the invention may also be applied to exhaust gases from any other source of combustion, including an external combustion engine, such as the Stirling engine or the Rankine cycle engine or to certain types of industrial combustion processes. Concerning IC engine exhaust gases, the subject matter of their formation in the combustion chamber has been thoroughly researched for over half a century (by Sir Harry R. Ricardo, among others). The question of exhaust gas inter-reaction and reaction under conditions of heat or catalytic action has over the preceding decade been the subject of the most major concentrated research the world has yet seen, with thousands of millions of dollars spent annually worldwide, and fruits of this activity have been set out in innumerable papers, reports by government agencies, journalistic articles, patent specifications, etc. It is felt that the science of exhaust gas reactions is also too well documented to be described here. A further reason for the omission is that the invention may apply to any IC engine including rotary, two-stroke and compression ignition, while chemical analysis must confine itself to a very particular specification and type of engine, fuel, material composition and ambient temperature of reactor and/or catalyst associated with it. It is known that the complex chemical reactions differ greatly with variation of these parameters, while perhaps still obeying the basic principles of the tri-component approach described. or the principles of other basic approaches such as passage of gas through a sequence of alternate oxidizing and reducing reactors (for example, the Questor system). Occasionally the sequence of the first two of the basic reactions described in section three is transposed, leaving HC to reduce NOx, depending on catalyst used. Mention should perhaps be made of an undesirable secondary reaction which has been causing some concern among environmentalists. In certain systems

employing first a strong catalytic reduction reactor and an oxidization reactor, ammonia has been formed with certain of the catalyst compositions employed. The nitrogen has reacted with the hydrogen present in the exhaust to produce ammonia: $2NO + 5H_2 = 2NH_3 + 2H_2O$. If the oxidization catalyst is downstream the ammonia is oxidized back to nitric oxide, so making the process of pollutant removal self-defeating. In the case of the present invention it is extremely unlikely that ammonia will be formed, for the reactor is intended to operate in the stoichiometric fuel/air mixture range normally without extra air. Similar considerations apply to the reformation of NOx.

As may be seen, the invention does not primarily relate to chemical reactions but rather to the provision of a more suitable environment for those reactions to take place in. For that reason it is proposed to describe in this section means of affecting or controlling reactions to a desired level, rather than the reactions themselves. The principal known reaction controlling means nearly all involve the provision of air or oxygen to reactor being an exception, the regulation or adjustment of the reaction taking place in the engine combustion volume, for example by variation of ignition or valve timing, the provision of EGR, the alteration of carburetor calibration, the enriching or weakening of the fuel mixture, etc. Such methods are all known and they and their effects on exhaust gas composition are fully documented.

It is proposed to provide an additional or alternative means for the regulation of engine combustion process, by allowing for the provision of two separate substances to the charge of ingoing gas, such as air. The first substance is the fuel, while the second substance may be a second fuel, a non-combustible agent or the latter mixed with fuel. The introduction of a second substance, continuously or otherwise, could measurably contribute toward engine power and/or improved exhaust emission and/or fuel economy. This latter aspect is relevant (it also relates directly to the first consideration) because of the "crises" concerning reduced availability of fossil fuels such as petroleum. The second substance may be introduced under, and assist in the

effectiveness of, certain running conditions such as sharp acceleration, high load or maximum power output. At such operating modes fuel consumption is greatly increased, but if the main fuel could be maintained at normal flow and the increased needs met by a second substance which is obtainable from non-fossil fuel sources, then a considerable saving of the main fuel is likely. Desirable as alternatives are, it is most probable that for many years to come the prime power of automotive units will be supplied by oil products. The second substance employed may be another fuel, such as alcohol or methanol which may be manufactured from such substances as waste paper, or it may be water in the form of liquid, vapor or gas, known since the turn of the century to give improved performance under certain conditions and tending to have an anti-knock effect, or in a preferred embodiment may consist of a mixture of the two.

The introduction of a second substance will affect the composition of the exhaust gases and emission control reactions. Water is nearly always present in excess quantity in exhaust gases, so the introduction of more water will not drastically affect reaction processes, although it can significantly reduce nitric oxides by cooling, if the water is introduced as liquid and absorbs latent heat in its conversion to steam in the high ambient temperature of the combustion volume. This cooling effect can be overcome if the water is introduced as steam. At very high temperatures water (and also products of combustions such as carbon dioxide and nitric oxides) tends to dissociate, producing $H_2$ and $O_2$. This dissociation involves the absorption of heat (which may be compensated for by the provision of heat in steam) which is not necessarily returned as cooling takes place and some $H_2$ and $O_2$ reforms to water. The provision of extra oxygen and hydrogen separately in the exhaust gas can in certain embodiments be of assistance in the reduction of pollutants, such as oxides of nitrogen. Concerning methanol, it can sometimes produce more power for a given volume than petroleum, due to its improved evaporation; but it might encourage knock and is preferably mixed with water. Water introduced as a liquid in the cylinder expanding to steam, or steam introduced under pressure, may greatly improve the volumetric

efficiency of an engine. It is felt that the benefits of the multiple substance supply to charge will increase in practice in proportion to combustion temperatures. It will be borne in mind that the invention is adaptable to all IC engines, including those likely to be developed in the future. Below are disclosed means for the introduction of two substances possibly simultaneously to an engine charge. In alternative embodiments more than two separate substances may be introduced. In addition to methanol, any other suitable hydrocarbon, for example ethanol, may be mixed with water. The introduction of water may be related to atmospheric humidity and regulated by a sensor.

In the section which follows are described means of introducing substances to an intake charge which do not involve the vaporization of fuel by gas velocity. Any of these means may be employed for the introduction of both the secondary substance and/or the main fuel to the charge. Additionally the secondary substance may be fed to a separated section of a carburetor processing the main fuel, this separated section only becoming operative under certain driving modes. The two substances may be gravity fed to intake area by separate reservoirs, or they may be fed from a combined reservoir, such as the float chamber illustrated schematically in cross-section in Figure 95, where a standard type float 261 moving vertically on spindle 262 to actuate shutoff lever 263 is positioned within a housing 264 containing volume for substance one, the housing 264 being concentric within outer housing 265 defining volume containing substance two and forming a barrier between the substances. The outer volume contains a hollow cylindrically shaped float chamber 266 slidably mounted on guides 267 to activate shutoff lever 268, fuel connections, shutoff valves, lid design and sealing all being according to conventional usage. A single, multiple-liquid containing float chamber assembly may be subdivided in any way, including by example the embodiment shown schematically in section in Figure 96, where one circular float chamber 269 and one crescent shaped float chamber 270 are mounted in separate volumes each on spindles 271. The multiple substances reservoirs have been shown of

circular cross-section. but may be of any suitable cross-section, including oval or rectangular, and of any shape elevationally.

In the case of compression ignition engines or other engines having cylinder or port primary fuel injection, the other substances may be supplied by means of additional injectors or any of the means described and alluded to above, or they may be introduced by compound injectors, that is by different passage systems in the same injector. The injection may be linked, that is injection of one substance will automatically cause the introduction of another, or the systems may operate independently of one another. Figure 97 shows by way of example a diagrammatic section where the primary fuel 272 is injected in the normal way at 273 by the lifting of nozzle 274 having a hollow central passage 275 linking with a secondary fuel gallery at 276 only when nozzle lift and consequently normal fuel injection is taking place. The secondary fuel is under continuous pressure and will therefore inject at 277 only when nozzle lift occurs. The proportion of normal to secondary fuel is determined by their respective pressures and the duration of degree of overlap between gallery and hollow passage. Figure 98 shows diagrammatically a compound injector having an inner nozzle 278 coaxial and within the outer 279, operating in the conventional mode with independent lift and injection capacity. This has the possible disadvantage of the long fuel travel down the hollow passage of the central nozzle. By way of example, a design involving a shorter central nozzle fuel travel from pressure reservoir to tip is shown schematically in cross-section in Figure 99 and in plan in Figure 99a where the nozzle assembly is viewed from the combustion volume. The central nozzle 280 operates in the conventional manner, moving vertically on its axis in the release of fuel, while the outer nozzle 281 moves coaxially on the first and in its seating in a rotational mode during the release of fuel. The rotational movement is imparted against the resistance of friction seals 282 by means of jets 283 terminating tangentially to diameter of nozzle. so imparting to it a twisting motion due to the force, and for the duration. of fuel injection. This will result in a slinging of fuel across the combustion volume in the

manner indicated at 284, in a similar manner to the action of some garden hoses. The injection of the outer nozzle is effected by means of a pressure wave in the coaxial and surrounding fuel chamber 285, which will depress one or more plungers 286 against spring 287 loading, and so by inward movement mate up fuel galleries to make connection and allow for fuel travel between the chamber 285 and jet 283 tip. The jet 283 has inappropriately been called such to distinguish it from nozzles proper as at 280 and 281. This slinging action imparted by rotational nozzle movement, the latter in turn imparted by the tangential direction of fuel spray, has considerable benefits over conventional injection systems. They operate in straight line distribution of fuel, while the snakelike shape formed by the spray of invention is of greater length, thereby lessening the chance of liquid deposition or combustion in chamber walls before atomization has taken place. The slinging action also tends to distribute the droplets of fuel through a greater volume of charge than the conventional unidirectional injection.

The rotary injector has been described in a composite embodiment, but in an alternative embodiment the rotary principle may be embodied in an injector handling a single substance. The rotatable member projecting into engine working volume may be of any configuration, and head configurations suited to rotatable injectors may also be embodied in fixed or non-rotatable head injectors. Rotation may be achieved by fuel injection velocity only, or by electrical action such as performable by solenoid or electric motor or magnet, or by flexible or fixed mechanical drive to injector. Rotation may be intermittent, continuous, or returnable, for example as when the head rotates during injection and is wholly or partly returned to its former position by spring or other action. Rotation may be achieved by any combination of the above means, as for example in an injector where a small electrical motor imparts rotational impetus insufficient normally to rotate head against bearing/seal friction loading, rotation only being achievable during substantially tangential injection, which provides additional rotational movement to overcome bearing friction. Mechanical or electrical rotation may be transmitted by means of a solid or hollow needle or tube or injector

nozzle seal which may be integral with rotating head or communicating with and/or driving it by means of splines, teeth, friction surfaces, etc. The needle/shaft/tube may simultaneously function as rotational drive and fuel release means by lift-off seat. In such case vertical movement may be actuated by conventional fluid pressure valve or by solenoid. If rotary motion is also solenoid actuated, one solenoid assembly may be employed to effect both motions simultaneously by means of suitable angling of solenoid action, as shown diagrammatically in Figure 166. Activation of electrical circuit causes shaft 800 to be pulled through one motion extent and direction indicated by arrow 801. Cessation of electrical circuit causes shaft to travel extent and direction shown by dotted arrow 802. Additionally, rotational motion of injector head means may be actuated by rotor or fan driven by fluid flow associated with engine function, as diagrammatically illustrated by way of example in Figure 147, described elsewhere.

The injector heads of the invention include configurations wherein fuel delivery means project into combustion volume at substantial angle to vertical injector axis, whether these rotate or not. The heads in a majority of configurations will be of solid material having formed within its passages for transmission of fuel. In alternative embodiments the heads have flexible elastomeric or spring action walls, so that initial increase in fuel pressure or arrival of fuel will cause head internal fuel transmission volume to expand or distend, remain distended during injection and following pressure cut off returned to normal position and cause residual fuel to be "wept" or expelled from head. In this or other embodiment of injection heads part or all of head may be of thin walled construction, and/or manufactured of thermally conductive material so that after pressure-actuated injection residual fluid in head is caused to evaporate or boil off. Such a feature will be useful in certain combustion engines to ensure continuation of combustion through a greater part of stroke, providing a more constant pressure type of engine operation. One projecting head assembly or multiple projecting head assemblies may be provided in association with one injection unit. The axis of rotation of injection head may be aligned in any relationship with the volume to which

injection is provided. For example, although injection and therefore axis of rotation will generally be envisaged as being in rough alignment with reciprocating motion of any engine piston, the axis of rotation may be substantially at right angles to reciprocal action of piston. As has been indicated, the rotational motion of head may be continuous, sporadic, jerk action, reciprocating (i.e., turning first in one direction, then in the opposite) and if continuous, constant or variable speed in the course of injection period and/or revolution. Any of these motions may be of speed or degree variable in relation to different modes of engine operation.

The intervention further comprises reciprocating, retractable and projectable and/or telescopic action injection heads. The reciprocating injection heads may move to and fro in fixed relationships to engine cycle or portion of it such as compression and/or expansion stroke. These entail the slidable mounting of a hollow member inside or outside of a hollow guide member of similar configuration, or of a multiplicity of such slidable members mounted about one another in nesting fashion, and may be fixed or movable (e.g., rotatable) in other planes. The slidable members may be straight or curved in elevational profile, and be of any convenient cross-section including circular, blade-like, cruciform, star-shaped, etc. The general retractable action may be incorporated in an injector for one or both of two significant reasons; to provide controlled fluid supply to working area far removed from injector base when cyclical motion of engine body portion permits (e.g., when piston is before say two-thirds of way up compression stroke), or to provide better fluid mixing or atomization generally. Fluid may be delivered through holes in end and/or other portion of slidable members communicating with interior hollow portion, and/or delivery may be effected by disposing holes of differing cross-section area, location, quantity, and/or alignment in adjacent members slidable about each other, so that in operation a controlled sequence of multiple fluid delivery is effected from hollow core of member(s) to working volume. The slidable or otherwise reciprocally moving member may have mounted in association with it a projecting or head portion, including those disclosed previously.

Reciprocal-type motion and rotational-type motion may be imparted to injector head by any means, movements being independent or linked. For example, as illustrated in Figure 167 member 803 communicating with injector head may be rotatably mounted on fixed sleeve or cam 804 of "hill and valley" profile to impart the combined motion referred to. Alternative solenoid assemblies operating in any manner including similarly to principles shown in Figure 166 may be employed to impart combined motion. Reciprocating and/or projecting/retracting motions may be imparted to injector head by any means, including those mentioned above, and/or by means of injection pressure extending or projecting head portion against say spring loading. In preferred embodiments, pre-injection pressure build-up will cause injector head portion to extend with some issue of fluid through injection apertures, with major injection taking place at considerably higher pressures once extension had been initiated, reduction of pressure causing cessation of injection and retraction of head portion. Alternatively, extension of head portion, say against spring loading, may be achieved by the combustion process itself, for example where portion of injector head defines a pre-combustion area or chamber of combustion engine. In such configurations the pressure of gases expanding in the pre-combustion chamber when firing commences causes the injector head portion to be "blown" or forced to a different portion say against spring action, and to return at any later period, including when pressures in main and pre-combustion chambers equalize.

To the knowledge of the applicant, other injectors involve fluid supply from a fixed point. As will be seen from later description, this leads to improved control of combustion process and/or flame spread in combustion engines. It also leads to a more uniform distribution of fluid in charge, which in combustion engines normally entails increase in efficiency and/or reduction in fuel consumption. It may not be readily apparent what a difference slewing the fluid through working volume will make. To illustrate this point better one may consider a garden hose with a given rate of water flow which one holds for a given period in a fixed position. Soon a large puddle

will form in one place with surrounding area relatively dry. If one held the hose with same flow-rate for same period but gave the hose light oscillating, flicking or stirring agitation, then the area of garden under consideration would receive an even spray of water with no formation of puddles. In a similar manner the slewing of fuel into a combustion charge would result in reduced fuel deposition on chamber walls, improved atomization, mixture standardization, and evenness of burning and would result in significant increases in engine efficiency.

A further feature of the invention is an injector assembly which partly defines volume suitable for commencement of combustion, or which causes such volume to be defined by the manner of injector assembly fitment to engine. The pre-combustion chamber may only be properly defined by fitment of injector, portion of which forms part of pre-combustion chamber wall. Alternatively the injector may have wall or shrouding assembly adjacent head which partly encloses pre-combustion chamber volume.

It is a further feature of the invention to provide a combined ignition and injector unit. Spark or arc ignition may be instigated by electrical bridge across terminals on the combined unit, or between one terminal mounted on the unit and another terminal mounted on or formed by other engine member, including chamber or pre-combustion chamber wall or valve, piston or rotor head, etc. The terminal(s) on the combined injector or injection unit may be of any configuration, including dome. L-shaped member, ring, including ring coaxial with unit axis and be of any convenient electrically conductive material, including metal and carbon. Ignition may be along current "cold" spark principles or along principles now under development which involve using a "hot" arc, including those systems referred to as plasma ignition wherein the arc causes a jet of super-heated gas to be expelled rapidly through an aperture to ignite a combustible mixture. In the case of the latter ignition system being incorporated in a combined ignition and injector unit, the ignition means, whether in singular or

plural form may be mounted adjacent to injection means, or the ignition means could be mounted coaxially with at least portion of injection means such as needle. In a preferred embodiment, the small chamber in which arcing and super-heating of gas occurs to provide plasma ignition is additionally provided with fuel supply means, so that the same chamber acts as source of plasma ignition and pre-combustion chamber. In another preferred embodiment, portion of ignition system such as needle acts as one terminal of an ignition system, including arc of plasma ignition system.

The following descriptions, read with reference to the diagrams where appropriate, show by way of example how features of the invention may be embodied. Figure 168 shows in elevational plan view an injector head capable of rotation, having three cranked hollow tubes 811 permitting fluid 810 issue through end hole. Figure 169 shows a similar arrangement, wherein multiple straight hollow tubes 812 each have multiple holes to permit fluid 810 issue. Figure 170 in elevational plan view shows a hollow disc 813 capable of rotational having one internal volume communicating with circumferential holes 814 permitting fluid 810 issue, arrangement of holes being shown in detail part end elevation Figure 171, the disc having coaxial with rotational axis another internal volume 815 capable of admitting passage of second fluid and which is closable by stem 816 mounted poppet valve 817. Figure 172 shows in cross-section during non-ignition period split disc 818 suitable for fixed as well as rotational applications, wherein the disc has flexible walls so that under pressure it assumes outline shown dotted at 819. Holes 820 permitting fluid issue are provided in communication with volume 821 between halves of disc, to which fluid can be supplied from passageways 822 in stem 823 or the central axial passage 824 closable by needle valve 825. In a preferred embodiment the split disc 818 is of thermally conductive material to cause fluid present in volume 821 during compression and/or combustion to tend to atomize, evaporate or boil. In a preferred embodiment in an internal combustion engine the injector provides a short burst of superheated steam via passage 824 during compression stroke, fuel is supplied under pressure via passages 822 about top dead

center of stroke, flushing out residual steam/water from volume 821 and an optional second short burst of pressurized superheated steam is admitted substantially during expansion stroke to flush out residual fuel and/or carbon and to provide additional pressure on piston. The flushing actions will assist in the prevention of deposits about the ends of holes 820. Figure 173 shows elevational plan view of injector head having looped hollow tube 826 of semi-spiral configuration, suitable for rotational and non-rotational application, with fluid 810, shown opposite injection holes. Although reciprocating, rotatable or otherwise movable members have been described in association with injector head assembly, the entire body portion of the injector including head may be so movable.

The art of mounting rotatable, reciprocal or slidable members is well known, these known techniques being readily employable in the construction and embodiments of the invention. In nearly all varieties of construction the fluid to be injected can be partly used as lubricant. By way of illustration there is shown in cross-section in Figure 174 a rotatable head 827 screw fixed to rotatable drive member 828, both being located by fixed injector body 829, with bearing surfaces 830 being lubricated by seepage from injection fluid volume 831 via a pressure-wave inhibiting ring 832, manufactured for example of ceramic fiber material.

Figure 175 shows elevationally and Figure 175a shows in sectional plan view a telescopic reciprocal or "lizard-tongue" action three part injector head assembly of blade-like cross-section. In Figure 175 it is shown solid in non-injecting position and dotted in fully extended position. The majority of holes for fluid 810 issue are in the long ends or sides of the blade-like sections 835, the latter extending against tension of wish-bone configuration leaf springs 833. Further holes 836 are provided to align with each other at certain stages during extension of the assembly.

Figure 176 shows lower portion of injector fitted to engine head or block 840 in such a way that a pre-combustion chamber 841 is formed to give access to main combustion chamber 842. Injector head 843 is movable rotationally and reciprocally, say by means of the device of Figure 167, from the position shown solid to that shown dotted at 844, and is mounted in fixed body portion of injector of non-conductive material such as ceramic. Conventional type spark terminals are shown at 845, with an alternative single terminal shown at 846 for providing spark to engine wall 847. Figure 177 shows a combined injector/ignitor having ceramic body portion forming shroud 848 defining pre-combustion volume 850 containing extensible needle injector head 849 having central end hole and controlled bearing weep to provide fluid 810 injection, plasma ignition means being provided at 851 to provide jet of superheated gas 852 during ignition. The entire injector of Figure 177 may be rotatable. Figure 178 shows a similar arrangement where electrically conductive shroud 848 is insulated from electrically conductive telescopic action injector needle head 853 by means of ceramic material 854, with ignition taking place by arc or spark between projective terminal 855 and needle head 853. Figure 179 shows a rotatable disc configuration injector head 856 in retracted position to partly mask pre-combustion chamber 841 from main combustion chamber 842. Ignition means are provided at 857, so that firing in chamber 841 will cause injector head to be blown to position 858 against spring loading (not shown).

It is a further aspect of the invention that the injector head portion be capable of reciprocal movement effectively to comprise a piston member. In a preferred embodiment this feature is used to provide a variable capacity pre-combustion chamber volume, as illustrated for example in Figure 176, where 860 shows in dotted outline alternative positions of injector head assembly. Optional sealing rings are provided at 861. Optionally, the movement of injector head and therefore of pre-combustion volume size may be variable while the engine is in operation, either manually or automatically, and be dependent on such factors as temperature, starting

condition, engine speed and/or load, intake charge pressure, atmospheric pressure, charge composition, fuel employed, etc. Such variable position piston or head assembly constructions are known in association with other devices and may be embodied in any appropriate manner. One way of carrying the invention into effect would be to bias by spring loading the injector toward its most retracted position against a rotatable cam operative against injector assembly base. Injector movement may be directed by any system of guides, channels, grooves, projections, depressions, ledges, cams, etc. Injector components may be of any suitable material, including ceramics, ceramic glasses, etc. Any injector head assembly of the invention may have reciprocal motion during each injection (to effect a slewing of injected fluid) and the degree of this reciprocation be made variable according to engine operation mode, say by means of cams capable of rotational and axial movement.

**FORM OF HOUSING**

Under this section it is proposed to describe various forms of housing wall construction, ways in which the reaction volume shape and association with engine block may be adapted to suit various types of engine configurations, how the housings may be subdivided into sections and how these sections may be affixed to each other, and ways in which the reactor housing may be associated with the inlet system and fuel supply of the engine.

Generally in the previous embodiments describing the internal face of the reactor housing, that exposed to the exhaust gases, has been regular. This may have the disadvantage, according to the nature of filamentary material deployed within the reactor, of tending to define a path of lesser resistance to the gas flow 300, as shown diagrammatically in Figure 100, where 301 is the housing, 302 the engine block, 303 say filamentary wool and 304 the less obstructive section between wool and housing. This

will result in too great a proportion of the gases travelling this path of lesser resistance rather than passing as intended fully through the filamentary material, with a result that some of the gases will not as fully inter-react as the system allows for. In order to mitigate this usually undesirable effect, the interior face of the housing may incorporate a series of depressions and/or projections, designed to break up gas flow adjacent to housing face and to direct as much of the gas inward towards the core of filamentary material proper. Figure 101 shows in diagrammatic elevation part of the inside face of a reactor housing, having a series of possibly alternative projections, with Figure 102 a corresponding section. By way of example, at 305 are shown a series of spaced straight ridges, while at 306 are curved intermeshing ridges and at 308 interconnecting ridges. At 309 are shown dimples or nipples, while at 310 are irregular projections of star-like or cruciform configurations. Figure 103 shows examples of how filamentary material fixing means may break up gas flow, with 311 a trench-like depression, 312 a projecting collar and 313 the ridges and troughs of earlier description. The internal face of the housing may further be waved as shown in diagrammatic part elevation in Figure 104 and in part section in Figure 105 showing a similar configuration where the waves are not continuous but form a succession of dune-like shapes. Both waves and dunes may be of regular cross-sectional configuration, as at 314 or may have a shallow slope facing the oncoming exhaust gases 300, and a sharp slope on the leeward side of the gas as at 315, or vice versa. In Figure 106 is shown how a ridge 316, optionally acting as filamentary retaining means, directs the flow of gas away from the junction between housing 301 and filamentary core 317, say of honeycomb configuration. Since the housing comprises at least partly insulating material there will be a large temperature drop between the inside face of the housing assembly and its outside face. Because of the high internal temperature of the reactor, perhaps in the 1100 to 1200 C. range, the temperature drop may not be sufficient to result in a surface temperature sufficiently low to prevent accidental burning by operating or service personnel. Largely to obviate this danger, the surface of the housing may be provided with protective ridges as at 318 in Figure 105 or nipples as at 319 in Figure 106. There will be further temperature

drop between surface proper and extremity of projection, but a much smaller hot surface is presented to accidental contact, thereby limiting heat absorption and degree of possibly burning.

The reactor housing may be incorporated with all or part of the inlet system of the engine, as illustrated diagrammatically by way of example in the case of a four cylinder engine in cross-section in Figure 107 and in elevational cross-section Figure 108 along component joint line 320, where 301 is the main reactor housing, 321 an intermember reactor housing, 322 exhaust gas exit, 325 carburetor assembly, 323 inlet manifold and 324 the outline of the exhaust ports. The principles illustrated above may be applied to the integral reactor and inlet housing for any configuration of engine cylinder bank. For ease of manufacture, the reactor assembly of Figure 107 has been made in two major components, which are fastened together in use in such a way as to facilitate the replacement of filamentary material. Jointing and division of components of this type, although not so shown, can be incorporated to the housing of any configuration, including those illustrated in this disclosure. They may be removably attached to each other in any manner, including the method of fasteners such as bolts placed in co-incident hollow load distributor ridges, as described earlier. Alternatively, the joints may be effected by spaced or continuous arcuate form back to back "L" members 326, as shown in part cross-section Figure 109, where the "L" members by means of interconnecting bolts 327, nuts 328 and wash plates 329, press the two components together, preferably at a joint having mating non-flat surfaces as at 330 to ensure proper location of components, separated by compressible material 331. In other embodiments, especially where the filamentary material is expected to last the life of the entire reactor assembly, it may be desirable to have the reactor fitted as a complete unit and remain effectively sealed, perhaps because the manufacturer wished to guarantee that factory settings were never tampered with. In such a case the various components of the reactor could be properly assembled and the jointing effected with a permanent adhesive or by a mastic jointing compound which would

bond adjacent surfaces together after the assembly had been fired or heat or chemically treated.

In the case of V-8 engines, costs may be saved and a better reaction environment created if both banks of the engine be made to discharge exhaust gases into a common central reactor 332, as illustrated in diagrammatic cross section in Figure 110. With a central exhaust collection point in an engine fitted to say a motor vehicle, some difficulty may be experienced in conducting the gases away to rear of the vehicle, due to restrictions of under-hood space. In a preferred embodiment, the reactor has one or more twisting exits 333, as shown by example partly in Figure 110 and in plan view Figure 111, and longitudinal section Figure 112 at M. The exhaust ports will not be too unequally spaced from one of the two gas exits, and the arrangement allows for twin exhaust pipe/muffling systems underneath the vehicle. The same twisting principle may be applied to a single exit from a reactor, with consequent reduction of valve and other duplications. In an alternative arrangement suited to "V" configuration engines, especially long ones of four or more cylinders per bank, it may be more suitable to have an end exhaust gas exit. If it were a disadvantage that some gases were to spend longer in the reaction volume than others, then a longitudinal gas exit pipe 335 may be disposed within the reactor, as shown in diagrammatic section Figure 113. By the careful arrangement of the taper of conical form of the pipe, and the displacement of its entry holes 336, the equal travel of exhaust gases 300 from port through filamentary material 337 to exit pipe 335 could be assured. Such a pipe may also be used in reactors of other forms.

It is intended that in some embodiments the housing of the reactor and/or inlet assembly assists in the supply of fuel, or of more than one different type of fuel to an engine. One of the most suitable materials for the construction of housing is ceramic, having low thermal conductivity. By controlling the thickness of material between reactor and any inlet and/or fuel provision system, one may accurately

determine the temperature of fuel and/or inlet gas in general or at a particular locality. This fact may be used to assist in the proper charging of the mixture in a number of ways, either during continuous use or under certain operating modes. One such method would be to provide for the vaporization of fuel by heat, rather than air velocity. An example is schematically illustrated in Figure 114, a cross-section part of a housing 339 incorporating two inlet manifolds 340 passing over the reaction volume 338. Between the manifolds and over the reaction volume, from which it is separated by a relatively thin wall 341, is a vaporization chamber 342, gravity fed with liquid fuel by means of a passage 343 within the housing communicating with a reservoir 344. From the vaporization chamber passages 345 lead to the manifold either directly or by means of multiple apertured hollow needle assemblies 346. In operation, the liquid fuel enters the vaporization chamber where it evaporates or boils due to its contact with heated wall 341. Because of the confined volume of the chamber, vapor or gas will be discharged through passages 345 into the inlet charge. Sufficient fuel will enter into the chamber and vapor formed to build up pressure, the degree of which is determined by minimum cross-sectional area of passages 345. This pressure will result in the reduction of liquid fuel entry to point where just enough enters to replace vapor escape and so maintain the pressure at an equilibrium. This balanced state is dependent on the precise design of chamber volume, floor area and temperature, gravity or pressure of fuel feed, size of fuel entry area, passage size and configurations. Such a system could supply fuel at rates proportionate to inlet gas volume flow rates and operating mode, because it is under pressure and fuel vapor flow rates would be sensitive to inlet vacuum. The effect of gas velocities could properly affect fuel flow rates, according to design of passage 345 and needles 346. In a preferred embodiment, illustrated in position in longitudinal section Figure 114a and detail section 114b. the needle 345 would have a hollow core 347 containing fuel vapor or gas and communicating with inlet charge flow 348 by roughly perpendicular fine passages 349 along its length, and larger passages 350 in the area of its rounded streamlined head 351. In operation, such a needle will exude a fuel vapor in rough proportion to inlet vacuum

through passages 349 and in rough proportion to gas velocity through passages 350. By careful design of the above and other features, proper mixture control can be achieved using heat vaporized fuel. The basic principles described can alternatively be used to provide and maintain at the right temperature other products to relate to, control or assist the engine combustion processes, such as for example steam or super-heated steam. These principles may be employed to supply one or more different substances to one engine, simultaneous or otherwise, and are preferably embodied in materials of low thermal conductivity better to maintain temperature, and to control these at specific locations by degree of exposure to and distance from heat source. The above may all further apply to substances supplied to engines only under specific driving modes.

The inlet section of the housing assembly may have disposed within it wicks to wholly or partly provide fuel or other substances to the engine charge. By way of example Figure 115 shows in cross-section and Figure 116 in longitudinal section a tubular wick 352 inside and against the face of the inlet portion of the housing 353. Fuel, gravity or otherwise fed, fills depressions in the housing comprising a main supply channel 354 and secondary distribution channels or grooves 355. The wick preferably has a progressively varying diameter to ensure contact with a greater proportion of incoming air or gas, and may work on either or both of two principles, namely that either the fibers absorb and carry the fuel, or they define capillary passages which transmit the fuel. If the former, the wick may have fibrous extension within or across the inlet, for instance in the form of gauze mesh. In general a wick will be sensitive to, and transmit fuel in proportion to, variations of vacuum or pressure. It is sometimes less suited to respond in the correct proportion to variations in gas velocity. Should the increase in fuel transmission with increase in velocity not be great enough, the wick may be arranged so that the fibers have a bias to lean toward the direction of gas flow 357, as shown at 358 in Figure 117, when engine is inactive. As gas velocity starts and increases, it will progressively force the fibers against bias to a more

perpendicular position, as shown at 359 in Figure 118, thereby exposing more fiber surface, and therefore more fuel transmitting surface to the airflow. The inverse of the foregoing may alternatively be applied. The wick need not be of circular section, but as shown by way of example in Figure 119, may have a segmented cross-section. Similarly it may be of greater length on the outer side of a curve, where the gas flows will be more intense and faster and so more effective, than on the inside as shown in Figure 120 at 360. As indicated diagrammatically in Figure 121, a wick 361 may transmit fuel at separate localities, or may transmit different, substantially non-mixing fuels 362. Different wicks transmitting substantially non-mixing substances may be used in association with or adjacent to each other within a single engine system.

The reactor constitutes a single unit from the engine block to exhaust pipe proper, notwithstanding that its volume may be divided into sections having differing catalytic effect. The cold start procedure has been described as effectively damming the reactor exit. In the gas of reactors having relatively large volume, the cold start damming may take place within the reactor, dividing it, as shown schematically in cross-section Figure 122 into a fore portion 364 and a rear portion 365 separated by dam 363.

The forms, contents and constructions of housing described in this and previous sections may all be employed in any combination and embodiment to provide a housing to treat, control or process in any manner incoming engine charge. It is known that engine efficiency and exhaust emission depend to a considerable degree on such factors as temperature, swirl, fuel atomization, etc., of engine charge. Previously most internal combustion engines have had charge supplied in the form of tubular columns passing through tubular manifold pipes. By passing charge through the housings of the invention much of the pulsing effect and critical tuning associated with conventional manifolding will be eliminated providing a smoother charge flow, especially during changes of operating mode. The provision of filamentary material inside a

charge housing can assist in improving turbulence, heat exchange, atomization of fuel, elimination of fuel condensations, etc. The charge housing may be formed similarly to the reaction housing disclosed in Section Four, with portion of charge treatment volume intruding into area normally taken up by cylinder head/engine block. Inlet ports may be formed of progressively varying cross-section to ensure smooth fluid flow between volume and main portion of port. Filamentary material may be provided anywhere in the charge treatment volume, but in preferred embodiments is in or adjacent to inlet port. The inlet port area, including adjacent and projecting into charge treatment volume, may have fluid distribution or flow controlling members such as or similar to those described in Figures 19 to 28. The fluid may proceed from charge treatment volume by non-parallel paths, for example similarly to the disclosure of Figures 17 and 18. Intermembers may be provided between charge treatment housing and engine body, along lines disclosed in Figure 6 and Figure 14, these being optionally of insulating material to maintain charge at ambient temperature. In the case of combustion engines, the housings, constructions, port arrangements, and contents of the invention may be applied only to process charge, or to process exhaust, or to both. In the latter case, charge housing may be opposite exhaust housing (as for example in "cross-flow" engines) or both housings may be mounted adjacently on the same engine side, either separately or in combination.

A principle advantage of using the housings for charge processing is that opportunity is provided for ultimately supplying a more uniform charge of fuel to each cylinder of a multi-cylinder engine. In preferred embodiments the housing will communicate with a plurality of inlet ports, so that fuel supply to charge before or in the housing will, provided proper turbulence or mixing has taken place, ensure reasonable equality of fuel supply to multiple ports. In the case of single carburetted engines, such equalities are usually similarly achievable with manifold systems, but in the case of injected engines the optimum balances are achievable by injecting into the charge treatment volume, either using conventional injectors, or the fuel delivery

systems of described elsewhere in this disclosure, or by using a hollow needle assembly similar to that disclosed herein but which forms a rail loop or series of such mounted substantially transversely to charge flow. A further considerable advantage of the invention is that it will provide improved inlet silencing. The above disclosure relates principally to combustion engines, but where relevant may be applied to any type of engine or pump.

By way of example there are shown in Figures 145 to 151 housing assemblies suited to controlling or processing charge of internal combustion engines, wherein 700 is main portion of housing, 701 charge processing volume, 702 direction of charge flow, 703 inlet port, 704 engine body or block/head portion. Figure 145 shows a single carburetor 705 mounted on housing 700 containing randomly disposed wire filamentary material 706, optionally supplied with oil to form oil-bath filter by means of wick 707 inset into upper inner surface of housing, fed from oil reservoir 708 covered by screw cap 709, there being disposed about inlet port large spiral flanges 710 cast integrally with metal cylinder head to provide pronounced swirl to gas in port and rapid heat transfer from head to warm up charge. Figure 146 shows fuel provided to charge by wick members 711 fed from channels/reservoirs 712 in upper portion of housing inter-connected to fuel supply through variable valve 713. For a given rate of fluid flow through 701, free fuel flows to wicks with valve 713 open will provide greater fuel absorption/richer mixture than with fuel flow to wicks restricted with valve 713 closed. In this embodiment it is desired to provide charge to combustion chamber as cold as possible, so an intermember 714 of thermally insulating material is provided between main housing 700 and engine 704. Between housing and engine filamentary material is disposed, in the form of multiple layers of gauze 732. Figure 147 shows a single injector distributing fuel to volume 707, here by twin rotating disc nozzles 719 mounted on injector body 715 giving continuous variable injection, with rotational motion actuated by fan 716 mounted about spindle 720 before injector in supporting bearing portion 717 of housing, rear of injector being supported by struts 718, fuel supply line

to injector being omitted for sake of clarity. Air cleansing is provided by a large trumpet-shaped paper filter 721. Optionally the rotational nozzles operate at differential pressure, one being active under conditions of boost. Cross-section Figure 148 and longitudinal section Figure 149 show diagrammatically alternative forms of hollow needle 735, hollow tubed loop 736 or hollow rail 737 fuel supply means, operative along the principles disclosed in this section. In this embodiment they communicate with and are fed from lower fuel passages 733, connected to fuel flow valve 734. Holes in hollow members are adjacent fuel "spray" shown at 738. Plan view Figure 150, longitudinal cross-section Figure 151, transverse cross-section Figure 152 show diagrammatically a combined charge and exhaust housing assembly where 722 is exhaust reaction volume, 723 exhaust ports, 724 direction of exhaust gas flow and 725 twin carburetors. In conventional engines, having inlet and exhaust mounted on the same side, the ports are substantially adjacent to one another. In an engine of the invention, the ports are disposed substantially above each other, as shown by way of example in diagrammatic elevation Figure 153, of a four cylinder engine, where 703 are inlet ports and 726 doubled exhaust ports. To obtain the desired head depth, the engine has substantially raised combustion chamber walls as shown by way of example in diagrammatic cross-section Figure 154, where the cylinder liner 727 seals against gasket 728 and block 729 seals against head 730 by means of gasket 731. The invention may further be incorporated in the form of central housing for V-formation engines described previously and illustrated in Figures 110 to 114, the gas flows obviously being reversed to those shown. The housing may be divided and joined in any manner, including those described in association with exhaust gas reaction volume housing.

A feature of the invention is the provision of a variable diameter charge intake throat. This may be used with any type of engine, but preferably forms charge entry point to the housing of the invention, especially if the housing is provided with fuel delivery means near entry point, as for example in Figure 147. Essentially the

variable throat comprises a stretched elastomeric tube about which is wound one or more tension members whose free ends once pulled effect a reduction in tube diameter. This will have the effect of increasing gas velocity through throat for a given engine speed, and is especially advantageous to effect proper atomization of fuel during slow running, idle to start up conditions. Section plan view Figure 155, cross-sectional view Figure 156 and detail Figure 157 show diagrammatically a stretched rubber throat 739 fixed within charge housing 740 by means of clamp rings 741, shown solid in open position. Wound externally about elastic throat 739 and mounted in lubricant 743 in guide channels 742 are multiple tension members 744 of nylon (shown in detail section Figure 157), whose ends are taken via pulleys 745 and wound about variable diameter cylinder 746 mounted adjacent to throat. In operation, rotation of cylinder causes tension member to effect a partial strangulation of throat, so reducing its diameter, as shown dotted in Figures 155 and 156. It is desirable that throat or membrane 739 when in the open position should be in significantly greater tension due to stretching in direction 747 than in direction 748, this differential ensuring throat remains open.

It is known that in gasoline engines best atomization of fuel usually occurs if charge is subjected to turbulence. This takes place to a degree when charge is flowing and substantially moving, which takes place in the greater part of the charge system. When this is least likely to occur is against an inlet valve when closed. Because the charge is moving during opening of the valve and suddenly stops on valve closure a wave or succession of waves of increasing and decreasing pressure is set up behind the closed valve. Although variation of pressure is present there is generally little turbulence or mixing during this period of valve closure, and because of the sudden cessation of charge forward momentum some liquid carried in suspension may be deposited on surfaces. In order to ensure maximum atomization of fuel while charge has lost forward momentum. the invention embodies the provision of flexible agitation or vibrating projections in the port area, preferably adjacent to valve. These

members may project from the valves which accelerate and decelerate rapidly and so will impart greater vibrating or oscillating motion to the members than would the port walls and guides, vibrating only to the degree determined by combustion taking place. Embodiments of agitation members are diagrammatically shown in cross-section in Figures 158 to 159, wherein 750 is inlet port, 751 engine wall, 752 inlet valve, 753 direction of main gas flow and 754 direction of oscillation. In Figure 158 there are shown a series of leaf- or blade-like members 755 projecting from inside of valve head, with wire or tubular members 756 projecting from port wall surface. Figure 159 shows an inlet valve 752 from whose stem project rod-or wire-like members 757. These have been inserted cold through holes in heated valve stem to form a press fit on equalization of temperature. Preferably the rod ends are then press flattened to give the oscillating member an oar-like configuration, as in 758. The projections need not be regularly spaced from each other, but may be concentrated in an area of port where greatest agitation is desired.

It is known that generally increases in combustion engine efficiency occur if the gas portion of the charge can be supplied to the combustion chamber at pressures greater than ambient pressure, this latter usually being less than atmospheric due to the sucking action of engine aspiration. Many types of force or pressure charging are known, including turbo- and super-charging. These all involve the engine undertaking the work of charge compression, which involves the expenditure of some of the work gained by charge compression. It is a feature of the invention to provide, in the case of motor vehicle engines, a means of effecting increase in charge pressures without extra work being directly expended by the engine. This is achieved by allowing part of the frontal aspect of the vehicle to act as an air ram, and to create in the engine compartment a controllable and/or partly sealable pressurized charge volume. As a means of regulating engine compartment pressure, an automatically or manually adjustable valve communicating to air bleed off passage and/or extractor may be provided. A schematic illustration of such an arrangement is shown in Figure 160, a

cross-section of front portion of a motor vehicle 760 where 759 is direction of normal travel, and 761 is direction of air-flow into and/or past vehicle when in motion. In the illustration air passed through main entry 762, through oil bath air cleaner 763, through radiator 764 into substantially sealable compartment 765 containing engine 766, there being provided valve 767 leading to bleed-off passage 768. Additional air enters from the frontal under-portion 769 beneath vehicle via scoops 770, this helping to reduce air pressure underneath the vehicle and so helping to improve road adhesion. The scoops may operate in progressive proportion to vehicle speed, not being operative below a certain speed, and increasing in effectiveness with increased speed. At low speed, therefore, the majority of air enters compartment 765 via entry 762, while at high speed a greater proportion would enter via scoops. This means under less ideal driving conditions the scoops would tend to be closed, tending to avoid intake of water, stones, dust, conditions where such are met tending to entail slower driving speeds. Variable scoops are illustrated in Figure 161 where scoop members 770 are shown mounted pivotally about 771 in closed position, the scoops having frontal lip 772 in projecting into air-stream 761 past vehicle and being biased to closed position by spring action indicated by arrow 773. In operation progressively increasing velocity of air acting against lip 772 will overcome resistance of spring action to open scoop. The valve member 767 may similarly be actuated in relation to vehicle velocity or by chamber 765 pressure or by a combination of both, as illustrated in Figure 161a wherein air flow 774 from engine compartment 765 is regulated by butterfly valve 775 sensitive to pressure against spring action 776, and by butterfly valve 777 activated by scoop 778 against spring pressure 779 which may be progressively increasable by mechanical or other linkage with lowering of vehicle gearing ratio, the valve being most ready to open in highest gear. Any other engine compartment bleed-off or pressure relief arrangements may be incorporated, including the provision of permanent or controllably variable gap between bonnet lid 780 and vehicle.

It has been said that the exhaust reaction volume may be provided with a division or barrier dividing it into portions. Similarly inlet charge treatment volume may be provided with any type of barrier, including those described elsewhere in relation to exhaust reaction volume and including any type of rotating member or fan, or rotor or turbine assembly. Alternatively or additionally, a rotating member may be in any other portion of exhaust reaction or inlet treatment volume, including at or adjacent to fluid entry or exit points. Means for mounting rotatable members having axes substantially in line of portion of fluid flow are known, especially in the art of turbine construction, but do not affect the principles of the invention. In the case of rotatable members mounted in exhaust systems, there should also not be any basic design problems, since many turbine assemblies are required to operate at temperatures in the same range as those likely to be found in the exhaust reactors of the invention. By way of diagrammatic illustration only there is shown in cross-section Figure 162 an exhaust treatment reactor volume 781 containing a fan or vaned rotor 782 integrally mounted on axial shaft 783 passing through special mounting formation 784 of housing 785 and supported on two bearings 786 in special drive housing 787, separated from exhaust reaction volume by means of lips 788 masking fibrous ceramic seals 789. In some embodiments the rotor and shaft are constructed of ceramic and other portions of drive mechanism of metal. Because of the differing expansion co-efficients of the two substances, it is desirable that they are mounted in association in special ways.

For example, there is shown in diagrammatic cross-section Figure 163 a ceramic shaft 791 carrying a fixably mounted bearing shell 790. Between them is a compressible barrier 792 say of fibrous ceramic, the internal circumference of shell carrying projections or splines mating up with splines on shaft 791. Rotatable members may be mounted in any arrangement in a housing. By way of illustration, there is shown in Figure 164 an exhaust reaction volume 722 having two rotors 793 mounted on shafts 794 having bearings at 795 and combined bearing and drive take off assembly at

796. Exhaust ports are indicated at 723, with line of inlet port passage indicated at 797. Figure 165 shows an inlet charge treatment housing having a single rotor 798 driven by shaft 794 mounted on bearing 795 and combined bearing and engine drive point assembly 799.

## MATERIALS AND MANUFACTURING METHODS

It is proposed to describe firstly those materials which are in general suitable for the high temperature and mechanical requirements of the invention, and then to describe materials particularly suitable to the filamentary matter in particular. Lastly various manufacturing methods will be described which are not especially well known or used as far as the inventor is aware, and which are considered especially suited to the production of the components of the invention. Material science is an immensely complex subject and has been rapidly expanding and developing during recent years, so for this reason it is proposed to give only an outline of the various material types and embodiments which may be used. The same considerations apply to a lesser extent to the question of manufacturing methods. Of course the invention in any of its embodiments may be made of any suitable material, including those not mentioned here and those which will be devised, discovered or developed in the future.

The more suitable materials for general use fall into three categories: metals, ceramics and glasses, and giant molecules generally known as polymers. Broadly, metals are ductile, resistant to thermal and mechanical shock, strong with progressive weakening with increase in temperature, tolerably resistant to abrasion and corrosion, in their refined and alloyed forms fairly resistant to temperature, and substantially in their elemental form. The other two categories do not have the same broad spectrum of advantageous qualities; ceramics and glasses, which are generally oxides or compounds of the half-way elements, have superior qualities in all respects

except ductility, resistance to shock and ease of working. However, because they are often very strong, more temperature resistant and generally much harder and abrasion/ corrosion resistant than metals, great efforts have been made over the last decades to overcome their disadvantages. New manufacturing processes have been devised, the mixes have been blended to increase resistance to shock and means of reinforcement developed. Concerning the polymers, these do not yet have the resistance to wear and temperature, or the hardness and strength of the materials, but they are beginning to be used as reinforcements and are also very suitable as insulating materials. They are capable of being the most elastomeric of the three groups and are useful for the manufacture of say the exhaust reservoir bellows of the invention, where temperatures are not as high as in the reactor. Polymers are being developed continuously; they are man-made and almost never occur freely in nature, and it is suspected that new super materials will be developed in the future by the polymerization of such metals as aluminum (next to silicon on the atomic scale) and some of the ceramic oxides. Many compounds do not fit clearly into one of these categories but lie in the area between.

The most suitable metals are the so called "super alloys,"- alloys based on nickel, chrome and/or cobalt with the addition of hardening elements including titanium, aluminum and refractory metals such as tantalum, tungsten, niobium and molybdenum. These super alloys tend to form stable oxide films at temperatures of over 700°C., giving good corrosion protection at ambient temperatures of around 1100°C. Examples include the Nimonic and Inconel range of alloys, with melting temperatures in the 1300° to 1500°C. range. At colder temperatures of up to 900°C. certain special stainless steels may also be used. All may be reinforced with ceramic, carbon or metal fibers such as molybdenum, beryllium, tungsten or tungsten plated cobalt, optionally surface activated with palladium chloride. In addition and especially where oxidizing reinforcement is not properly protected by the matrix, the metal may be face hardened. Non metal fibers or whiskers (often fibers grown as single crystals) such as sapphire-aluminum oxide, alumina, asbestos, graphite, boron or borides and

other ceramics or glasses may also act as reinforcing materials, as can certain flexible ceramic fibers. Materials, including those used as filamentary matter, may be coated with ceramic by vapor deposition techniques.

Because of their greater hardness and higher temperature resistance, ceramics are the most suitable materials for situations where mechanical and thermal stock loadings are not critical. In the case of the invention, they are especially suited to the manufacture of the housings, intermembers and port linings because of their generally low thermal conductivity. Suitable material include ceramics such as alumina-silicate, magnetite, cordierite, olivine, fosterite, graphite, silicon nitride; glass ceramics including such as lithium aluminum silicate, cordierite glass ceramic, "shrunken" glasses such as borosilicate and composites such as sialones, refractory borides, boron carbide, boron silicide, boron intride, etc. If thermal conductivity is desired, beryllium oxide and silicon carbide may be considered. These ceramics or glasses may be fiber or whisker reinforced with much the same material as metals, including carbon fiber, boron fiber, with alumina fibers constituting a practical reinforcement, especially in a high-alumina matrix (the expansion coefficients are the same). In fact it is the very high alumina content ceramics which today might be considered overall the most suited and most available to be used in the invention generally. The ceramic or glass used in the invention generally may be surface hardened or treated in certain applications, as can metals and often using the same or similar materials, including the metal borides such as of titanium, zirconium and chromium, silicon. etc.

The filamentary material may be made of metals, preferably smoothed and rounded to avoid undue corrosion, or of ceramics or glasses. Other materials which may in molten state be particularly suitable once they are in full commercial production are boron filaments, either of pure boron or compounds or composites such as boron-silica, boron carbide, boron-tungsten, titanium diboride tungsten, etc. The material, especially if ceramic, may easily and conveniently be in the form of wool or

fibers, and many ceramic wool or blanket type materials are today manufactured commercially, usually of alumina-silicate, and could readily be adapted to the invention. Such ceramic wool could also be used as a jointing material either alone or as a matrix for a more elastomeric material such as a polymer resin. The material may either be such to have catalytic effect, as in the case of many metals, or have a catalyst mounted or coated on the basic material, such as ceramic. The techniques of applying catalysts to ceramic matter are extremely complex, usually secret and covered by patent protection, but part of current industrial art, with many manufacturers producing catalysts applied to ceramic substrate commercially, and are not directly related to the invention.

High temperature lubricants will be necessary for moving parts, and these may comprise boron nitride, graphite, silicone fluids and greases, molybdenum compounds, etc. For perhaps the less direct mechanical applications, polymers may be employed. Silicones have already been mentioned as being suitable in rubber form for the expansible bellows of the reservoir of the invention, and may also be used structurally in harder, resinous form. Resins suitable include those of the phenolic family (e.g., polytetrafluoroethelyne) and boron containing epoxy resins. Other polymers suitable are for example the boranes, such as decaborane silicones containing uncarborane and other silicon-boron groups. These polymers may be reinforced with any whisker or fiber, including those mentioned above.

Wool, especially if of ceramic material, is often made by extruding or extracting fine jets of molten matter in a bath of cold fluid, usually liquid, a process which has been referred to previously as a fluid collision technique because of the force required and the rapid cooling on contact with the fluid. In a preferred embodiment, hot liquid filamentary material is injected through fine apertures, possibly arranged in the disposement of exhaust port layouts, into a restricted volume containing cold fluid which is of corresponding shape to reactor housing, the liquid on cooling forming into a

wool mass of generally the shape to fit into reactor housing. If the wool or fibers are too linear in configuration, then the cooling liquid may be agitated say in a twisting irregular motion preferably by impeller forced into a cooling reservoir through an aperture corresponding to the exhaust gas exit. In Figure 123 is shown schematically such a layout, where the molten material 370 injected under pressure through small apertures 371 into a reservoir 372 containing cold fluid 373 agitated by propellers 374, the resultant fibers formed indicated at 375.

The complex shapes that the filamentary material may comprise may be manufactured by a reversal process, whereby the forms of the intended passages are made up in material A, about which the filamentary material B is formed. Subsequently material A is dissolved or leached out in a suitable substance such as acid or water, leaving the material B only in the intended form. Such methods are known and suited to ceramic manufacture.

The materials may be formed by any of the current techniques now known, including slip forming, molding, pressing, stamping, sintering, extruding, etc. The isostatic pressing of powders is one of the more suitable means of manufacturing in ceramic the possibly complex shapes of the reactor housings, providing sufficient hydraulic pressure is available for the relatively large sizes of the objects. Pressing usually takes place on a male mandrel, which can accurately be made to the desired form. If the internal form entails difficulty of removal of product, then the male mandrel may be an elastomeric housing filled with an incompressible effectively fluid material such as liquid or powder or grains, these being removed after forming so that the mandrel may be collapsed inwardly. Figure 124 shows schematically in cross-section such an arrangement, where 375 is the base plate, 376 is the elastomeric collapsible male mandrel filled with fine sand 377, and surrounding fluid 378 under pressure exerts force via expansible outer membrane 379 on the powder 380 to form the desired product. In operation, the mandrel is filled with sand via closable passage 381. the

compression membrane 379 is fitted over the base and powder injected into the volume between membrane and mandrel through closable passages 382, preferably under pressure to properly fill it. The assembly is then placed in fluid, which is subsequently subjected to violent pressure, causing the compression of powder by means of the flexible membrane 379. The pressure is sometimes effected by explosion or detonation. After pressing has taken place, the membrane is removed, the sand 377 inside the mandrel is extracted, the mandrel 376 collapsed, say by application of vacuum through passage 381, and the pressed object removed. In many instances the external surface has to be machined to attain the right shape, sometimes because of inaccurate manufacture. The control of form of outer casing, and of proper even powder 380 dispersal before pressing can be improved in the following manner. The elastic outer membrane may have wall thickness deliberately varied at certain points, so that when powder filling under pressure and inevitable expansion takes place, the thinner (therefore less strong) sections expand further than the thicker thereby causing a corresponding projection in the object to be formed. By accurately controlling the rate of expansibility of different sections of outer membrane relative to each other, by means of variation of wall thickness, provision of stiffening ribs externally, etc., and also the precise amount of powder and pressure under which supplied, a form can be pressed accurately, with neither face needing post machining. Just prior to pressing the filled assembly may be subjected to agitation or vibration, to ensure the uniform distribution of powder 380 at even pressure. The pressing technique has been described with inner forming member non-compressible in operation but in an alternative the outer former may be non-compressible, with the inner being the membrane adjacent to working fluid.

## ENERGY RECLAMATION

It has been seen that the invention entails a means of treating exhaust gases to high emission standards without involving the use of the energy consuming devices many other systems need. Such devices include the variation from stoichiometric fuel/air mixture ratios, variation of spark and valve timing from optimum, the provision of EGR and extra air, etc. Apart from in normal configurations eliminating nearly all of these, the invention lends itself to the provision of exceptionally efficient gas flows, certainly improved over those through the majority of alternate emission systems, and in some refined embodiments more efficient than those through the exhaust manifold systems of untreated engines. Further, the cold start means here disclosed, by properly conserving and using the heat produced by engine combustion, eliminates alternative energy consuming cold start expedients such as electrical heating, heavy fuel enrichment, heavy air input, etc. As has been mentioned earlier, it is considered perhaps the most important advantage of the invention that it offers an exceptional scope for fuel conservation.

However, the principles of the invention relate also to methods of saving substantial further energy, by means of deriving further mechanical work from the combustion of a given amount of fuel and/or by means of the provision of energy storage in accumulators, so as to compensate for the stop/go nature of most vehicle operation. One such method is to raise the ambient temperature in the reactor to substantially over the 950°C. to 1200°C. range, thereby further assisting the desired process of reaction, by the substantial increase of temperature in the combustion volume thereby increasing the thermal efficiency of the engine. Another method involves the extraction of heat from the area of, or adjacent to at least the rear of the reactor to provide further work. Further, the invention may be used in association with means of converting the flow of exhaust gas into mechanical energy.

In order to both raise reactor temperatures to assist the desired exhaust gas reactions, and to raise the ambient temperatures in the combustion volume in order to increase thermodynamic efficiency, it is proposed to eliminate conventional cooling in an engine desired for continuous running, that is to eliminate heat dispersed from combustion chamber walls by means of liquid pumped through engine block jackets to a heat exchanger, or by means of cooling fins and usual associated air blower. It is intended to construct the engine to operate continuously in an uncooled state, so that it might be used to power for example, generating plant, light cars and trucks, heavy goods vehicles, locomotives, marine vessels including supertankers, etc. To this end, the uncooled engine preferably uses the internal combustion cycles although the principle of the invention may also be applied for example to engines operating on the Rankine or Stirling cycles.

In a conventional internal combustion engine (IC Engine), the rapid burning of the combustion charge in the confined space of the combustion volume produces expansion and heat. The expansion drives the piston and consequently engine while the heat product of the cycle is almost wholly unused - in fact considered undesirable since efforts are made to dissipate it as effectively as possible, by means of conduction through cylinder walls and head to cooling system. Other heat is collected by the lubrication system to be often dissipated by oil radiators, sump cooling fins, etc.

Let it be assumed that in a particular water-cooled engine the energy produced by the combustion or fuel is distributed as 32% going to useful work on the piston, 28% carried away by cooling water and 40% dissipated by exhaust gases and general radiation. If the heat loss to water jacket can be eliminated, about 5% to 6% will be theoretically converted to useful work on the piston, bringing the percentage of total energy converted to work up by 4% (allowing for losses due to increase of specific heat and dissociation at higher temperatures) to about 36%, corresponding to an engine power increase of 12½%. With the elimination of cooling system mechanical

losses, a further increase on the original figure, of about 4% - 6% can be expected, bringing the total power saving to say between 16% and 19%. Since water heat loss was eliminated with 4% out of 28% total energy converted to work, the remaining heat, 24% can only be carried away by the exhaust gases and general radiation, bring this figure from 40% to 64%, an increase of 60%. If the general radiation can be eliminated in the same way as the water heat loss, and if this made up say 19% out of the original 40% total with exhaust, then some of this, say 2% would be converted to useful work, corresponding to power increase of 5% over original output, and the remaining 8% total energy would be carried by exhaust gases. These figures suggest that the provision of an uncooled engine would involve power increases between 12% and 25%, and increase of heat carried away by exhaust gases of 40% - 80%. Allowing for various factors, this suggests an increase of exhaust gas temperatures in the port from between 850°C. and 1100°C. to somewhere between 1000°C. and 1500°C. suggesting temperatures within the reactor of the invention of between 1100°C. and 1600°C. If only water jacket heat losses were eliminated, the port temperatures would likely be in the 950°C. to 1400°C. range, with reactor temperatures in the 1050°C. to 1500°C. range. Combustion volume surface temperature would rise from those in conventional engines say between 150°C. and 300°C. currently to between 250°C. and 450°C. An uncooled engine could not therefore be constructed entirely in conventional materials, and alternative embodiments are described below. It must be borne in mind that a projected power increase of 12% to 25% without increase in fuel consumption (none is required) must be considered a very valuable saving considering today's energy climate. Allowing for margins of error, an even 10% fuel saving - a given power is necessary for a certain engine function, so fuel consumption would be saved rather than power increased - would make critical difference to the oil needs and political situation of a highly mobile country such as the U.S.A.

The uncooled engine may consist of components constructed of any material suited to the environment found in the engine location in which the com-

ponent is used. In a preferred embodiment heat loss is eliminated by omission of cooling and construction of engine/cylinder block at least partly of materials having heat insulation properties, such as ceramic. Types of the latter material are among the few able to withstand the ambient temperatures found in certain sections of the uncooled engine, such as the exhaust port area. As has been mentioned in the previous section. ceramics are generally hard and more abrasion resistant than metals, and may sometimes be stronger, especially if reinforced. It is feasible, according to today's technology, that virtually all the components of the engine may be made of ceramic, including such items as main bearings, connecting rods, etc. However, in a more practical embodiment, the moving parts are of metal of a construction and type conforming to current practice, with the possible exception of the exhaust valve. Figure 132 shows by way of example schematic cross-section of an uncooled engine having a ceramic engine block 400, a ceramic cylinder block 401, camshaft 402, valve 403, port 404, cam cover 405, sump cover 406, carburetor 407, crankshaft 408, connecting rod 409, piston 410 and combustion volume 411. All moving parts are metal except the ceramic exhaust port, seating detail of which is shown in Figure 133 where valve 403 seats against compressible seal 412, optionally lubricated by passage 413, in cylinder block 401. Figure 134 shows an alternative detail, where valve 403 seats against compressible seal 412, optionally lubricated by passage 413, in cylinder block 401. Figure 134 shows an alternative detail, where valve 403 seats against ring 414 slidably mounted in groove 415 containing between ring and groove floor 416 a compressible cushion 417, lubricated by optional passage 413, the cushion forcing ring slightly outward when valve is lifted. If necessary the compressible material may be bounded to groove floor and/or ring member, to better prevent the latter leaving the groove. The compressible member may be constructed out of ceramic fiber and serves as a shock absorber at valve closure, ceramic not being as ductile and resistant to certain types of mechanical shock as metal. The piston is of material preferably a heat resistant alloy such as nickel-chrome, having ceramic piston rings, to ensure that the mating surfaces have corresponding wear coefficients. Finning at the bottom of the

piston gives some cooling to the crank volume, which may be part cooled through the sump. The piston could equally be manufactured of ceramic or other suitable non-metal. Lubrication would be by any suitable substance, including those mentioned elsewhere. If lubrication were such as to easily pick up particles of say ceramic, which would damage softer metal bearing surfaces, then metal piston rings might be used to ensure that wear produces powder of the softer material, metal. Such an engine would be considerably lighter than conventional units, especially if construction used light, high alumina content ceramics. Considering also the elimination of cooling mechanics plus fluid, the overall large weight reduction would further contribute to fuel savings, where the uncooled engine is used in vehicles. The construction of engine blocks at least partly in insulating material would greatly assist in the reduction of noise and vibration, thereby providing additional social benefit. Ceramics is defined in the preamble to the recitation of claims, and embodiments are described elsewhere. Gaskets between ceramic components may be of ceramic such as asbestos mat.

Ceramic engine/cylinder block construction leads to the introduction of several beneficial features. Passages and chambers to transmit substances such as fuel, air, steam, water, etc., may be incorporated within the block(s), perhaps to embody the principles outlined in other sections, in a manner to ensure their transmission at the desired temperature and/or pressure according to distance of passage from combustion volume. Similarly, electrical circuits can be incorporated in the body of the block, since ceramic can be an electrical insulator. Such circuits may connect to electrodes or points, say of carbon, in the cylinder head to produce a spark without the need for conventional plug. High voltages may be employed to give larger sparks, say arcing through substantial dimensions of the combustion volume, without fear of these large sparks shorting against the block. Such circuits could be incorporated by pouring molten metal into passages already formed in the manufactured ceramic block.

A reactor assembly mounted to an internal combustion engine may have incorporated within or adjacent to reaction volume (whether associated with conventional or uncooled engine), a heat exchanger, so that the heat of the exhaust gases may be used to heat the working fluid of an alternative engine cycle expending work on either another engine or on the original (which thereby becomes a composite engine), or to heat fluid communicating with an electrical generator or an accumulator. Figure 135 shows diagrammatically such a configuration, where an engine 418 having exhaust ports 419 discharges exhaust gases 420 past finned members 421 having hollow passages shown dotted 422 communicating with lower linking passage 423 and upper linking passage 424 formed in reactor housing 425 and having access to, respectively, fluid entry means 426 and fluid exit means 427. Such heat exchangers could be made of a material having high conductivity, including ceramics such as silicon nitride or metals such as the nickel alloys, which may be such as to have catalytic effect. The heat exchanger may effectively constitute filamentary material. Alternatively, the heat exchangers may be placed elsewhere in the exhaust system of an engine, including just behind reactor assembly.

The heat exchanger may be part of an engine cycle putting work into an accumulator, a second engine and/or the first engine. It may pool work with the first engine by means of mechanical linkage, or by the partial integration of the two engine cycles to produce work on common components, such as piston or crankshaft, the latter embodiment constituting a composite engine. If the heat exchanger were part of a separate mechanical power unit, it could be coupled to the first unit by direct drive. If the latter is used in an automotive application, the power requirements of the stop/start nature of operation may not always conform with the more constant outputs the regular supply of exhaust heat and possible working fluid pressure will provide from the second power unit. Therefore the second unit may be connected to both the first unit and an accumulator by means of a differential, as illustrated diagrammatically in Figure 136, where 428 is the first engine, 429 the reactor/heat exchanger assembly, 430 the

second engine, 431 the differential and 432 the accumulator. Drive shafts are provided at 433, and the accumulator may optionally be linked by passage 434 to first engine 428. The accumulator may comprise a fan compressing fluid such as air to be stored in an associated reservoir, in which case the bleed off of fluid to first engine 428 under certain operating modes such as acceleration may result in improved performance or fuel economy (see other sections).

The heat exchanger may be used to heat fluid including air, other gases, water to steam, steam or superheated steam. These fluids may be used as outlined elsewhere, i.e., to provide addition to the charge substantially during induction stroke of the first engine, or it may be used to power a second engine, perhaps coupled to the first engine as above, or it might be applied to operate the exhaust and/or compression strokes of the first engine, thereby embodying a composite engine, or it may be employed to operate some pistons of a composite engine having other pistons operating on the internal combustion cycle. In the latter case the pistons may operate on the same crankshaft, which in a preferred embodiment is divided by a say multiple dog-toothed clutch to eliminate inter-reaction of torsional vibration between crankshaft sections. By way of example, Figure 137 shows diagrammatically an arrangement whereby crankshaft section 435, driven by four IC operative pistons, is connected to crankshaft section 436, driven by alternatively two steam cycle or Stirling cycle operative pistons, by means of a multiple toothed dog clutch shown in cross-section at 437 and in elevation at 438. If the two operating cycles employed are such that optimum efficiency occurs for each at differing revolution rates, then the crankshaft sections may be connected by gears 438a of suitable ratio, as shown in diagrammatic plan Figure 138 and section Figure 139 where 439 is the IC powered piston and shown dotted outline alternate powered piston 440, with 441 axes from gudgeon pin centers to crankshaft centers. If the fluid is required to act on the piston common to an IC engine system, such a piston is preferably of T-shaped configuration, as shown diagrammatically in section Figure 140, where a piston having hollow head 450 reinforced by flanges 451

attached to hollow stem 452 in slidably mounted in a cylinder 453 by means of piston rings 453 and bearing 454 notched to accommodate piston flanges, and the piston separating IC operative combustion volume 455 and alternate combustion and/or expansion volume 456. Piston stem communicates to crankshaft 457 via big end bearing 458, connecting rod 459 and gudgeon pin 460 according to known practice. The fluid of the alternate system may be further cooled (heat will have been given up if expansion has taken place) by passing through a heat exchanger, say taking heat from fluid to assist conversion of such heat into electrical energy or mechanical energy. By way of example, a layout suitable for the employment of the Stirling hot gas principles as alternate cycle is shown in Figure 141, where S and T are chambers having pistons linked by common crankshaft, the reactor/heat exchanger assembly shown at 461 and the heat disposal exchanger mentioned above at 462. Cold gas enters chamber S along path 463 to be compressed and travel under pressure via path 464 to reactor 461 where it is heated to then travel via path 465 to chamber T, where it provides work on expansion, then travelling at low pressure via path 466 to cooler 462, to thence repeat the cycle. Here one piston and chamber effects only compression while another only expansion. In an alternative system illustrated in Figure 142 each piston/chamber assembly operates alternatively on compression and on expansion, considering only the alternative engine cycle.

The heat exchanger may comprise part of a turbine engine cycle as shown diagrammatically by way of example in Figure 143, where an IC engine 467 having exhaust gas 468 passing through reactor 469 across heat exchanger 470 to drive fan 471, which is linked by shaft 472 to drive turbine compressor 473 to pass compressed turbine working fluid 474 via passages 475 through reactor heat exchangers 470, allowing expansion of turbine working fluid to occur. A fan associated with the reactor of the invention may drive a compressor used for any suitable purpose, including the provision of a compressed fluid to an accumulator and the provision of boost to engine inlet charge.

Figure 180 shows a schematic arrangement for a gas turbine engine mounted in association with an internal combustion engine 900 in such a manner that the exhaust gas from engine 900 provides the means of heating the gases of turbine engine 901, wherein working gas passes in direction of arrow 902 through intake 903, low compression stage 904, high compression stage 905, heating stage 906, turbine stage 907 and exhaust stage 908. Exhaust gas in alternative embodiments either flows through heat exchangers in stage 906, being optionally compressed beforehand by separate compressor 910. A combination of both systems may be used, as may supplementary fuel combustion system in stage 906, as shown at 911. Such combinations of internal combustion engine and turbine engine are suitable for aircraft, railed vehicle and large trucks, for example, where exhaust through 908 may be used to provide extra motive power. The schematic arrangement of Figure 180 may be used to provide a combined steam turbine and internal combustion engine. Figure 181 shows in schematic section a turbine assembly 913 semi-integral with exhaust gas housing 912 for an internal combustion engine 914 so that two or more gas flows are substantially coaxial or parallel, with Figure 182 being a part section through Z. Rotor assembly 915 has three separate coaxial bands of blade assemblies of configuration differing from each other, and is mounted on a bearing assembly 916, with power take-off means of teeth at 917 driving gear 918 and shaft 919. Exhaust gases drive rotor which causes air to be sucked in at 920 across oil cooler 921 and steam condenser 922 associated with steam system powering compression and exhaust stroke of internal combustion engine 914, this steam being heated in heat exchanger 929. The arrangement of rotor blades at 923 causes the gas to be compressed and pass through heat exchanger 924, heat being absorbed from exhaust gases 925 by means of thermally conductive wall 926. to expand and drive rotor 915 by means of vane 927 and stator 928 assembly then mingling with automotive exhaust gases. Ducting means are provided at 930 to take some compressed gas to internal combustion inlet charge, thereby effectively providing turbo-charged internal combustion portion of engine.

An uncooled engine may be construed in any manner. If components such as ceramic are used, they will probably be relatively more difficult and expensive to produce in large pieces than in smaller ones. For this reason the engine is preferably made up of smaller units which are assembled during construction of the engine. Diagrammatic elevation Figure 183 shows by way of example an engine composed of multiple pieces 930 built up round combustion chambers shown dotted 931 and held together by means of bolts 932 in tension. Figure 184 shows an embodiment of engine having double head construction, with upper head 933 admitting inlet charge at port 934 and expelling exhaust at port 935 (both shown dotted) for internal combustion, and with lower head 938 having inlet port 936 and outlet port 937 for steam cycle. In assembly the engine is built up about piston 939 and combustion chamber wall 940 of sleeve-like configuration, having seals or gaskets at 941, by means of spacer or aligner blocks 942 and tension bolts 943. Poppet valves 944 and cam assemblies 945 are provided to regulate fluid flows. Heat transfer 962 (in the form of steam condenser) may take place between ports 937 and 934, and between ports 935 and 936 (say in the form of steam heater or water boiler). The two head construction may also be used in engines with both sides of piston operative in the internal combustion mode. Figure 185 shows a means of fixing a mechanical assembly 946 to a block or engine portion 947 of insulating material such as ceramic. A bolt 948 having load distributor head 949 is passed through a hole 947 and spaced from it by a compressible interlayer 950, of say fibrous ceramic. If the bolt has greater coefficient of expansion than the block portion 947 then a strong spring 951 and washer 952 may be provided to keep contact between assembly 946 and block 947 at constant pressure with differential expansion of bolt and block. In an embodiment a metal sleeve 953 forming combustion chamber wall has a compressible interlayer between it and insulation block. The metal liner permits use of conventional piston assemblies, including sleeve valves 949a as in Figure 186. Figure 187 shows a piston suitable for operation inside say a ceramic liner 954, being of composite construction having main metal head and body portion 955 seated inside ceramic integral skirt and ring assembly 956 and spaced from it by compressible

interlayer 957. Fixing is made by pivotally mounted spring clips 958. Figure 188 shows a combustion chamber/piston assembly similar to that of Figure 184, but having a hollow mushroom-shaped piston head 959 reciprocating between domed heads, the upper head 960 having ball valves 961 similar to those described elsewhere.

The problems of likely differential expansion between metals of conventional engine construction and the insulating materials (such as ceramic) of the invention can easily be overcome by intelligent detailing and design. For example in Figure 188 there is shown a metal poppet valve 970 mounted in a metal guide 971. Between it and ceramic block 972 is disposed a thin sleeve of compressible and slightly stretchable material, such as fibrous ceramic. The guide with sleeve is fitted to the block when the latter is at very much higher temperature than the guide. When temperatures equalize to ambient a tight fit will ensue, as when the engine is cold. When the engine is warm, the relatively greater co-efficient of expansion of the metal will ensure that the guide is an even tighter fit in the block. By using this and other techniques, an engine can be constructed of partly metal, partly ceramic and partly insulating material.

The above features and those of the ensuing sections illustrate by way of example the many ways an uncooled engine may be constructed. Any type of piston or valve may be used in an uncooled engine and the engine portions may be assembled in any manner. The features of the uncooled engine have been described mainly in relation to internal combustion engines, although they are suited to and may be applied to any type of combustion engine, including for example steam and Stirling engines.

The features relating to heat exchangers may be embodied in any type of engine, including conventionally cooled engines. Where appropriate, features described herein may be applied to pumps. By "uncooled" is meant engines having restricted or no cooling, compared to general current production engine practice and includes

engines with partial cooling. It is to be emphasized that the various features and embodiments of the invention may be used in any appropriate combination or arrangement. Where diagrams or embodiments are described, these are always by way of example and/or illustration of the principles of the invention. Further, it is considered that any of the separate features of this complete disclosure comprise independent inventions.

In conclusion it is to be emphasized that the various features and embodiments of the invention may be used in any appropriate combination or arrangement.

In the following recitation of claims, "filamentary material" shall be defined as portions of interconnected material which allow the passage of gases therethrough and induce turbulence and mixing by changing the directions of travel of portions of gas relative to one another, the inter-connection being integral, continuous, intermeshing, interfitting or abutting, this definition applying to the material within the reactor as a whole and not to particular portions of it. By "ceramic" is meant ceramic in the widest sense, encompassing materials such as glass, glass ceramic, shrunken or recrystallized glass or ceramic, etc., and shall refer to the base or matrix material, irrespective of whether other materials are present as additives or reinforcement.

## CERAMIC ADIABATIC INTERNAL COMBUSTION ENGINE (CAICE) EMBODIMENTS

### 1. Configuration Fundamentals

The concepts disclosed herein relate to improved configurations of internal combustion engines, especially though not exclusively to those made of ceramic materials and without cooling. Mostly two cycle engines are described but the ideas can be applied also to four cycle engines. Methods are disclosed for increasing the efficiency and/or exhaust gas temperatures of such engines. Partly using the improved configurations mentioned above, ways are presented for integrating IC with turbine or jet engine cycles.

An important concept involves the substitution of conventional engine elements generally in compression by tensile elements. For example, a push rod is replaced by a "pull wire." The arrangement is illustrated diagrammatically in Figure 240 with camshaft 1256 actuating rocker arm 1257 fixed at pivot 1258 which, via tensile member 1259, activates rocker 1260 anchored at pivot 1261 which activates valve 1262 and spring 1263. It is clear that the use of tensile members permits greater freedom in location of cam and valve mechanism, since the line of force need no longer be a straight path. By way of example tensile element 1259 is shown routed clear of another engine element 1264 by means of wheel, roller or bearing 1265. The rocker arrangement of Figure 240 can be eliminated as shown in Figure 241 by attaching the tensile member 1259 to a movable cage 1266 surrounding the cam 1267, the cage having a cam follower 1268 (shown by way of example as a roller bearing) and guide 1269 (shown by way of example as a flange slidable in a slot, the latter not illustrated) for defining follower movement relative to cam in the direction indicated by arrow 1270.

A preferred embodiment of the CAICE is illustrated schematically in Figure 189 and consists of a piston 1001 oscillating between two combustion chambers 1002 at each end of a cylinder 1003 closed by two heads 1004 with a crankshaft 1006 outboard each head, the piston being connected by tensile member 1007 to both crankshafts. Optionally, the crankshaft will also function as a camshaft, actuating valves and optionally providing fuel delivery. The liquid elements for the charge may be delivered to the combustion chambers under pressures and temperatures higher than normal in conventional engines. The cylinder is at least partially surrounded by an exhaust gas processing volume 1008 with exhaust gas being conducted to the volume by alternate paths 1005 and 1009. Intake to the combustion chamber is via the crankcase. Surrounding the engine is a heavily insulated casing 1010. This configuration is suitable for four and two stroke embodiments, consuming fuel ranging from gasoline and similar lightweight fuels through diesel and heavier oil fuels to coal and other slurries or powders. Any engine lubrication bearing system may be employed, but optionally either gas or roller needle bearings are used, perhaps with water or other liquids, in the case of water preferably when the components are of ceramic material. This is further described in Subsection Three. In the case of air bearings the crank assembly is preferably so designed that its air bearings at least partially operate at a pressure equivalent to the charge pressure of forced induction in the case of turbocharged, supercharged or force-aspirated engines; in the case of two stroke engines the preferred arrangement is to exhaust gases via ports about the center of the cylinder. In the two cycle form illustrated schematically in Figure 242, pressurized air is ducted via crankcase 1275 and valve 1276 actuated optionally by combined crankshaft/camshaft 1277 to combustion chamber 1288 (injection system not shown) displacing exhaust gas which exit the chamber via ports 1289 to exhaust gas processing volume 1290. In another example Figure 243, the cylinder module 1271 is linked to a single crankshaft 1272 by tensile elements 1273 routed about guides/bearings/rollers/wheels 1274.

The layout described above may be arranged in multiple cylinder form in a flat configuration shown in plan Figure 190 longitudinal section Figure 191 cross-section Figure 192 where five cylinders and ten combustion chambers are arranged about two crankshafts connected at one end to the transmission 1011 and optionally mechanically linked by it, and at the other end driving ancillary systems 1275 and optionally linked by member 1012. Figures 190 through 192 have been dimensioned in terms of unit d, in this case and being both the bore and the stroke of the piston. In an alternative configuration shown in schematic longitudinal section Figure 193 and cross-section 194, a double row ten cylinder engine is shown. Obviously, any number of rows and cylinders can be combined between two crankshafts, since it is only necessary to lengthen the tensile elements. In Figure 195 and 196 schematic cross-section of a four row engine of eighteen cylinders and thirty-six combustion chambers is shown where tensile members 1013 and 1014 are of unequal length. Either separate cam-shafts or more elaborate valve/fuel activation or linkages are required to provide valve actuation or fuel delivery for engines having three or more rows of cylinders and two crankshafts. Alternatively more than two crankshafts can be employed as shown diagrammatically in cross-section Figure 197 and longitudinal section Figure 198, in the case of a six row forty-two cylinder, eighty-four combustion chamber engine. It will be noted that these configurations are most practical if the engines are uncooled or adiabatic. If the tensile members are replaced by connecting rods a single crankshaft may be used, as shown diagrammatically in Figure 199 for a two row engine, having a single combined crank/camshaft 1015 and two camshafts 1016, various valve actuation rods 1276.

The basic cylinder modules may be combined to form a "ring" engine with the interior space optionally used for a turbine or ram jet engine to form a compound engine having a single revolving system. Schematic sections Figures 244 and 245 show four modules 1277 linked by common crankshafts 1278, with hot exhaust gases 1280 providing at least partial energy for the ramjet or turbine 1279, either directly or via

heat exchangers (not shown). The work from the IC portion of the engine may be used conventionally, may power the compressor of the turbine portion or may, as shown schematically at 1281, drive a fan, propeller, archimedes screw to provide thrust, either through air or water.

## 2. General Design Analysis

The generalized design objective is to arrive at an engine having greater power to weight, power to bulk and efficiency than equivalent contemporary units. In the present case this is achieved by three principal means: the rearrangement of the IC engine components into a more compact and simple configuration, the drastic reduction of reciprocating masses, and the virtual elimination of heat loss from the system.

It is obvious that the CAICE configuration concept and the tensile link between the crank and piston concepts are interrelated, and together provide significant advantages. Substitution of the heavy connecting rod and its bearing at the piston by the much lighter tensile member entails that the crank can only be pulled, not pushed, and so necessitates a rearrangement of IC parts. With two combustion chambers acting on one piston less loads are transferred through the crank permitting lighter construction. This is especially true in the case of two-stroke engines, where virtually only net work and therefore loads are transferred to the crank. (Part of the work of expansion is transferred through the piston to provide most of the work of compression.) If the tensile link is used, the desirable slack will generally cause the piston to "float" toward the end of the first chamber's expansion stroke, a transition ensuing after combustion expansion causing the piston to pull one crankshaft and subsequently being pulled by the other crankshaft to effect final compression of the second chamber. A significant portion of the loads of piston deceleration will be taken up by the compressing charge and will not be transferred to the crankshaft, permitting lighter construc-

tion. Because of the constant line of the tensile member between heads, the piston is much less subject to side loads and torque, simplifying piston bearing and seal design. The arrangement of the exhaust processing volume adjacent to the cylinder eliminates heat loss from the cylinder walls to outside the system. If the volume is properly insulated exhaust temperatures will more closely approach mean combustion chamber gas temperatures, reducing thermal stress on the cylinder. Likewise the piston has two opposing work faces, and consequently will have shallower temperature gradients than conventional pistons. In the two stroke embodiment cold charge enters the low maximum compression end of the combustion volume thereby cooling it, while hot exhaust gases exit the cold minimum compression end thereby heating it, tending to even out the temperature gradients of the combustion chamber surfaces. Because these arrangements substantially reduce thermal gradients and consequently stresses it will be easier to manufacture the components in a wider variety of ceramic materials, which generally have less tolerance to thermal shock than metals. It is generally understood that engine efficiency increases proportional to charge temperature gradients and to a lesser degree with increase in compression/ratio, and that power to bulk and power to mass ratios increase proportionately to engine speed - provided that these increases are not partially absorbed by higher friction and pumping losses, and that combustion efficiency is constant within the speed range considered. It is an objective of these designs to provide an environment where combustion temperatures, compression/ratios and engine speeds all higher than in present units can be successfully and efficiently employed. The higher combustion temperatures will tend to produce hotter exhaust gases, leading to improved emissions control and usually a greater heat sink for waste heat recovery systems, which will therefore produce more work, and generally lead to greater system efficiencies. All the above would suggest that in the case of high performance engines, carburetor or manifold injected fuel delivery should be discarded in favor of direct injection into either cylinder or pre-combustion chamber so providing more controllable combustion and reducing the risk of pre-detonation.

It is now generally accepted that the more efficient engines of the future will be force aspirated, usually by turbo- or supercharging, and that most two stroke designs require some form of forced aspiration. Accepting that work must be expended into compressing the charge (the efficiencies gained by improved aspiration more than offsetting the work required), the present designs seek to use any such compressed environment to provide some portion of the work required for gas bearings, which is one reason aspiration can be via the crankshaft. Both the sliding interface between tensile member and head and the interface between piston and cylinder will preferably employ some form of gas bearing, probably a combination of high-pressure blow-by and/or water-generated steam bearing, described later in Subsection Three. This means that oil pumping losses (plus bulk, weight, cost and unreliability of such equipment) can all be eliminated, as can the heat dissipation of the conventional lubrication system. For practical purposes all friction losses can be eliminated, since the friction produced by gas turbulence in bearing clearances of a few microns' depth is negligible in relation to loads carried. Preferably inter-cooling is eliminated also. The consequent loss of mass of charge is offset by the higher charge temperature differential, but most importantly the pumping losses, waste heat dissipation, complexity, bulk, weight and cost of intercooling systems are eliminated. As noted earlier, important engine design objectives are simplicity and viable cost. So far we have an engine in which coolant and lubricant pumping losses as well as friction losses have been virtually eliminated. These are substantial on modern engines, especially high performance diesels, so this would suggest a proportionate increase of efficiency resulting from the elimination of these losses. There has also been virtually no heat loss whatsoever, assuming both crankcase and exhaust volume housing have theoretical maximum insulation. Heat dissipation through the head is of course transferred back to the charge. Since the difference between ambient and combustion charge temperature has been increased, there should be a proportional increase in efficiency. If it is desirable to increase combustion temperatures still further (the only physical limit being the temperature resistance of the combustion chamber materials), the compression/ratio

can be increased, providing further increase in efficiency. With increase in temperature the tendency for charge products to disassociate increases. Generally this is not desired and can be compensated for by increases in compression/ratios, since pressure inhibits disassociation. Because some of the heat is produced by combustion, increasing the compression/ratio will have a proportionally greater effect on absolute pressure compared to absolute temperature. Additionally water in some form may be introduced to the combustion process, which will have the effect of reducing temperature and increasing pressure, as described in more detail elsewhere. Due to either increased temperatures and/or pressures, efficiencies will be higher with the new engine.

An important feature of the present engine designs is the significant reduction of reciprocating masses, firstly by the elimination of the usually heavy connecting rod and its piston bearing assembly, secondly by the substitution of steels by ceramic materials of between 30% and 40% of the weight of steel, thirdly by the reduction of most of the rocker and push rod mechanisms of conventional engines. It is estimated that reciprocating masses could be reduced to end up weighing as little as 10% to 20% of conventional practice. Ignoring valve actuation let it be assumed that only a 75% reduction is achieved on the piston/crank system. If the stresses caused by the reciprocating masses increase roughly as the square of the increase in engine speed (see Note 1) then reducing the reciprocating masses by 75% will either permit double present engine speeds with the same stress limits, or a twofold reduction in stress limits. The strength of construction of an engine (and therefore its weight) is directly proportional to the required stress tolerance. In other words the present engine designs permit lighter construction with consequent weight savings and vehicle system efficiency, and higher engine speeds. Excluding mechanical (friction and pumping) losses and assuming combustion efficiency is constant, power to bulk and weight ratios increase proportionately to engine speed, as noted earlier. However, in these designs there are virtually no mechanical losses so in many cases the only practical limit to

higher speed is maintenance of combustion efficiency (reciprocating stresses being drastically reduced).

The current state of the art appears to indicate that with force aspirated engines efficient combustion can be maintained up to around 200 revolutions per second (12,000 r.p.m.) for gasoline engines and around 100 r.p.s. (6,000 r.p.m.) for direct injection or diesel engines, the limiting factors being the time taken for combustion to be initiated and once initiated to be properly completed and, in the case of direct injection engines, by the time taken to distribute the fuel throughout the combustion chamber. Both of the first two processes can be hastened by increased pressure, putting the constituents of combustion in closer proximity to each other, and by increased temperature. Therefore if compression/ratios are increased or water added to provide combustion pressures higher than those prevalent now, then a corresponding increase in usable engine speed is likely. The delay time may also be reduced or eliminated by delivering the liquid parts of the charge into the combustion chamber at greatly elevated temperatures and pressures, so that it vaporizes immediately on entering the chamber. Then the kinetic energy imparted to the mass of droplets during injection would have to be such that it would carry the fuel in a largely gaseous state to the desired regions of the chamber. In this mode the injection process could have some of the features of stratified charge, or plasma ignition in main combustion volumes. In comparison with a solid connecting rod, the effect of the tensile crank design will be to delay the piston at each end of the cylinder and hasten its passage between the ends. (See Subsection Three.) This delay at each end also implies that engine speed can be raised relative to conventional engines for giving combustion parameters. Taking into account piston delay, increased compression/ratios and combustion chamber temperatures, the delivery of fuel under high temperature and pressure, one might suppose that engine speed limits for a given efficiency of combustion could at least double. With additional new injector designs and layouts, speed

limits in direct injected engines might increase up to three or four fold, that is diesel speed limits might be in the 250 to 400 r.p.s. range.

For some reason three dimensional camshafts have not been generally introduced into today's engines. This seems inexplicable, since the cost of imparting axial motion to a camshaft is small and the benefits are great. These include providing variable valve lift and dwell, providing an optional and variable secondary opening to the combustion chamber for charge bleed off (providing a variable effective compression/ratio engine, or a means of improving two stroke charge purity if hot exhaust gases are adjacent the opening), providing variable ignition timing, providing variable fuel delivery actuation. In the case of engines with a large speed range, the variation in optimum settings for valve lift, dwell, ignition timing, etc., becomes greater, suggesting that three dimensional camshafts are desirable in the new engine designs. In those designs where it is proposed to integrate camshaft with crankshaft, it would be feasible to provide the crankshaft with three dimensional actuation if tensile members are used, and relatively easy to embody with gas main bearings.

## 3. Constructional Details

The design feature which is probably least affected by prior art is the tensile link between piston and crank. The various options and parameters are more complex than is immediately apparent. In the twin crankshaft layout described previously it is not possible to maintain a constant length between piston and crank. as shown diagrammatically in Figure 200 wherein 1100 shows equal synchronized crankshaft centers with throw of radius r rotating in the same direction 1101, shows piston 1102 and head/cylinder module 1103 of constant dimension k. solid line 1104 representing tensile member when the piston is in the middle of the cylinder and line 1105 the tensile members when the piston is at the end of the cylinder. In the latter position it will be seen that if crank centers are placed 3r length on piston axis outboard of

module, the total tensile length between crankshafts is 2r + 4r + k = 6r + k. When the piston is in the center the tensile member dimension is hypotenuse of right angle triangle base a-c plus hypotenuse of right angle triangle base df plus k. Since the bases total 6r and since the hypotenuses must be longer than the bases, it follows that the distance between the cranks is longest when the piston is in the middle of the cylinder. Since the components need always be linked, the length of tensile member is that required to accommodate the piston in or around the middle of the ~~piston,~~ cylinder, meaning that there will be slack in the tensile system when the piston is towards the ends of the cylinder. This slack is an important feature of the design of tensile crank engines and is described in more detail later. So far symmetrical situations have been considered - the same parameters apply to both of the combustion chambers of the piston. If the angles of the cranks are not synchronous, then asymmetrical conditions in the combustion chamber may be obtained, as shown schematically in Figure 201; tensile members are shown in alternative configurations 1106, 1107. Two optional tensile configurations aa and bb are shown when the piston 1102 is in the center of the cylinder travelling in direction 1108 to compression chamber 1109. When the optionally linked cranks have travelled through 180° tensile parameters have changed with respect to the identical piston now in compression relationship for combustion chamber 1110. (Obviously the cranks will complete revolution if the tensile members have the required amount of slack.) In order to better understand the principles of tensile crank design only symmetrical layouts will be considered from here on. The tensile link may be wholly of some flexible material, or may partly comprise a rod, as shown schematically at a and b in Figure 202. In both examples a constant portion of the tensile element is parallel to piston movement, in one case it is fixed relative to crank centers, in the other reciprocates. Here the cranks are shown turning in the same direction and the free portions of the tensile element angled at 180° or less to one another. Not shown, but equally possible, is to have the cranks turn in opposite directions to one another, thereby maintaining the free tensile portions at a more or less constant 180° to one another. In Figure 203 an arrangement for differential pivots for each half of the cycle is shown,

which will cause the piston to be off cylinder center when the cranks are 90° off BDC/TDC, so permitting differential piston speeds during cycle phases. For example, the piston could be moving faster during the main portion of the compression stroke compared with during the main portion of the expansion stroke or vice-versa.

Figure 204 shows at (a) and (b) how the configuration can be used for four stroke and two stroke engines respectively, with intake 1111, compression 1112, expansion 1113, exhaust 1114. In the case of the two cycle only net loads are transferred to crank; in the case of the four cycle alternately net and gross loads are taken up, suggesting that for a given number of cylinders the two stroke will be much smoother running. The configuration improves two stroke smoothness over conventional systems more than that of four cycle engines.

Referring back to Figure 200, in order to enable the crank to turn through the 90° to BDC/TDC, the free tensile halves will have equal slack at BDC/TDC. By enlarging crank movement radius, the slack toward TDC will be decreased and the slack at BDC increased by a slightly greater amount. Reducing the crank radius to less than that of piston movement reverses the process - there is most slack at TDC. It is also obvious, Figure 200, that the greater the distance from head to crank center in proportion to crank radius, the less slack required in the system. It is preferable to have little or no slack at TDC, since the piston as it approaches TDC may need to be pulled there by the crank to complete the compression and subsequently, as expansion takes place, the loads must be transferred as quickly as possible to the same crank. On the other hand, toward BDC all the useful work of expansion will have been completed, so a taut tensile member is not required. In practice, to enable the tensile member to be taut at TDC the crank movement diameter will have to be around 5/4 to 8/7 of piston movement, depending on design details. The ratio can be reduced for equal crank centers by employing the configuration of Figure 205.

Optionally, the engine may be so designed as to permit increased compression/ratio with increase in speed. For start up and low to moderate speed the arrangement described above is employed, the piston is pulled by a crank to a "designed" compression/ratio position and on expansion the piston in turn pulls that same crank. Before the piston has been pulled to complete compression it has been slowed down because its kinetic energy and the work done on it in the other combustion chamber by the last stages of expansion is less than that required to complete compression. (During this slowing down period the slack may be transferred from one free tensile half to the other; except for transition phases one tensile half is always taut and the other slack.) However as the engine speeds up the kinetic energy of the piston becomes greater to the point where when at the designed compression/ratio the work effected in and by the piston has equalled the work required for compression. As the piston speeds up further the work on it and by it, exceeds that required for the "designed" compression. Since the piston is not restrained other than by the compressed gas, it will compress the gas beyond the "designed" ratio. As piston speeds increase and compression/ratio climbs more kinetic energy is required, which is derived from the extra work obtained by burning a fixed mass of fluid at higher pressure and temperature. The prime benefits of increased compression/ratio with increased engine speed would be the shorter required combustion time, due both to increased pressure and the increased temperature resulting from higher pressures. This may or may not affect the disassociation processes going on during expansion. Generally these are encouraged by higher temperature and inhibited by higher pressure. Temperature and compression/ratio do not increase proportionately since the temperature is the result of pressure and combustion combined. (See Note 6.)

It is obvious that the deceleration of the piston has to be controlled relative to variation of engine speed to ensure that all slack is taken up in the relevant free tensile half close to TDC, and that the excess of crank rotational speed over speed of tensile half movement is small as tautness is attained, to as far as possible eliminate

shock loads on the tensile member. In the case of variable compression designs it is also desirable that tautness is attained at an angle of crank rotation before the loads of expansion can begin to be efficiently transferred to the crank. This control can be provided in the first place by designing the mass of the reciprocating parts to suit the desired engine speed range, and by varying the timing and quantity of fuel delivered before TDC. Optionally water, water/methanol or similar substances can be introduced before TDC to provide sudden increases in pressure at critical periods and/or to control too-rapid temperature rise. It is assumed that in some engines it will be desirable to have the greatest possible engine speed because power to weight ratio is important (e.g., aircraft applications), so the objective of the variable compression concept is not so much to increase efficiency (in some embodiments it might decrease) as to facilitate proper combustion in short time intervals. An interesting feature of the variable compression engine is that once the "design" compression/ratio has been exceeded, the masses of the reciprocating parts (other than valve and fuel systems) exert no loads on the crank. Therefore the traditional limitation to engine speeds in medium and large diesels is completely removed. As has been shown, the tensile crank design reduced reciprocating mass, as did the substitution of ceramics for steels, enabling much lighter engine construction to be employed. The variable compression concept removes reciprocating mass loads altogether at higher speeds.

The crankshaft itself may be manufactured along conventional lines and may be of any material, including ceramic. Non-conventional configurations may also be used including the built-up configurations shown schematically in Figure 206, wherein center bearing tubes 1115 and big end bearing tubes 1116 are mounted in compression by axial tensile fasteners 1117 between discs 1118 which act as crank throws. These discs may be so formed as to both permit maximum bearing size and allow the circumferential area to act as a cam, as shown in cross-section by way of example in Figure 207, where two shaped discs 1119 having precisely machined surface cam profiles 1120 for valve cam follower 1121 actuation and fuel delivery cam follower

1122 actuation. The discs are interconnected by tensile fastener 1123 and inner crank bearing cylinder shell 1124 having precisely machined ends, each disc being similarly fastened to an inner main bearing cylinder shell 1125. Outer main bearing cylinder shells 1126 are attached to engine structure. Outer crank bearing (big end bearing) cylinder shells 1127 are attached to crank connecting rod or tensile member 1135. The present layout is shown having gas bearings where the largest bearing areas are desirable, but of course roller or needle bearings may also be employed. The technology of both gas and needle roller bearings in ceramic and other bearings is well understood and not itself a novel feature. If the gas bearings required greater than ambient gas pressure, then gas passages 1128 communicating with a central gas reservoir may duct gases to apertures 1129 at the bearing surfaces. In an alternative arrangement suited to ceramic materials and high crankcase temperatures (say around 450°K and over) the passages may contain water under pressure, which on leaving the apertures will instantly turn to steam, so providing gas under pressure in the relatively close tolerance (sometimes 1-3 microns) of the gas bearing. Optionally the centers of the inner bearings cylinders may be filled with water - if the subsequent possible increase in crankshaft vibration is acceptable - to provide, together with likely counter-balances, some kind of flywheel effect. In crankshafts having few throws the gas or liquid may be pulsed, to provide maximum pressures at moments of greatest loading. Instead of the apertures, a combination of apertures and wicks may be provided as shown diagrammatically in longitudinal and cross-section in Figures 208, 209 where a wick 1130 is disposed at maximum loading area 1131 to more evenly distribute the liquid delivered under pressure via passages 1132 and apertures 1133. In arrangements described elsewhere, the slack in the tensile element may optionally be taken up by a fluid spring so that tautening of the tensile element causes fluid to be delivered to the bearings. The crank of Figure 207 is shown having lateral or axial motion permitting the cam followers to be actuated to varying degree by the progressively shaped cam profile as the crankshaft is moved indirection 1134. Here it is assumed that the link between piston and crank 1135 is not laterally movable, entailing larger inner bearing

cylinders or shells than outer ones. Water lubrication is cited as an example; in fact any suitable liquid under pressure may be used, whether or not it changes to a gaseous phase in the bearing gap.

A way of linking crank to a piston and rod assembly is by a tensile link, preloaded to always absorb slack in the system, using for example spring steel. Figures 210 to 213 show such a spring steel link 1136 in tension under load, biased to open to position shown at 1137 when all load is removed, wherein Figure 210 is a sectional elevation, Figure 211 a plan view, Figure 212 a detail section taken at b, Figure 213 a detail section of the components at c. U-shaped cross-sections of the tensile link at b and c permit bending and therefore lateral movement of the crankshaft as shown exaggerated at 1138. The flatter cross-section at the spring or fluid reservoir at 1139 permits bending to take up slack, as shown exaggeratedly at 1140. Two spring actions are indicated here, the biased spring steel and the spring/reservoir at 1139, although in fact only one is needed. The device shown at (a) is a shock absorber consisting of two roller 1141 linked by stiff low-extension springs 1142 in association with a compressible mat 1143 between spring steel loop 1144 and outer bearing shell 1145. Figure 213 shows an enlarged section of the joint between the tensile member and the end 1148A of the rod of a piston/rod assembly, where the wedge-shaped split ends 1146 of one tensile half are seated in a shallow conical depression 1148 in the rod end and located by collar 1147. The fluid reservoir is indicated schematically only, its volume not necessarily being to scale. Variation in stiffness of springing will affect the acceleration and deceleration of the piston during transfer of slack from one tensile half to the other.

Another method of linking the crank to the piston is by a flexible tensile element such as cable, rope, yarn, etc. One design is shown in Figures 214 and 215, here with a hammerhead rod/piston assembly 1149. A compressible fluid reservoir 1150 is linked to outer bearing shell 1151 from fluid supply reservoir 1158 by fluid

supply line 1152 with non-return valve 1153 and by delivery line 1154 with non-return valve 1155 to deliver fluid to bearing at 1159 via passages 1160. Twin tensile cables 1157 pass through a bell mouth 1156 to be wrapped round the shell, with ends 1162 press or adhesive mounted. Similarly, the cables are attached through bell mouths 1163 about the detachable hammerhead 1164. They may pass through the piston/rod assembly (not shown). The hollow rod 1165 has openings permitting the passage of charge at 1168 which is moved within the hollow space by the rapid reciprocal motion of the piston/rod assembly. The head is attached to the rod by screw threads 1166. Figure 216 shows a single cable mounted to a constant diameter rod tip 1167 of a rod/ piston assembly, where the cable enters through a split bell mouth 1169, passes through the rod to be wound about it then re-enters the rod to pass through to the other end.

There are many methods of attaching the piston to the tensile elements. Figure 217 for example shows a single cable passing through the cylinder head 1170 guided by asymmetrical crank revolution roller guides 1171 and crank lateral movement roller guides 1172. The cable is passed through cast-in passages 1174 provided in the integral piston 1173, wrapped about the circumference and then passed through the piston again. Optional voids 1175 have been provided in this piston. Figure 218 shows a similar arrangement in a tri-component piston, where the piston crowns 1176 are screw threaded to each other by means of a central cylinder or drum 1177 round which the cable is wrapped. A compressible sleeve 1178 is provided to project the cable against abrasion and to act as a shock absorber. Figure 219 shows an open skirted three-component piston, where the crowns 1179 are screw threaded to each other by means of smaller central cylinder 1180. Figure 220 shows an open skirted piston/rod assembly, where the rod 1181 is hollow and continuous, the piston 1182 having reinforcing flanges 1184 and is press fitted to the rod and attached either by the tightness of the fit (achievable by inserting the cooled rod in the heated piston) and/or by plugging a volume bisected by the component joint line as at 1183, or by a combination of both. The hollow rod may house a continuous tensile member 1185. Figure

221 shows a tri-component closed-skirted piston assembly 1186 assembled about two independent rods 1187. Compressible material is provided at 1188 to provide small movement of the piston on the rod for shock absorbing purposes and at 1189 to provide a thread lock. Hollow passages 1190 may communicate with the interior of the piston, to carry fluid, preferably gas, through the piston in direction 1191 for cooling or other purposes. Figures 222 and 223 show arrangements equivalent to those of Figures 216 and 218 except that twin cables are provided.

The tensile member may pass through the head in a number of ways. In rod/piston assemblies bearing surface must be provided where the rod passes through the head to take up the angled loads caused by crank rotation. In the case of cable assemblies these can be taken up by rollers, as for example in Figure 217. Figure 224 shows the rod 1192 which is reinforced during extreme crank angles 1193 by a sleeve 1194, which may be movable in direction 1195 and may provide fuel delivery. Normal piston/rod movement range is shown dotted at 1196. The sleeve has a cutout 1197 (shown in plan view in Figure 225) to accommodate this movement range. Where the tensile member has to take lateral crank rotational loads, bearing may be by high pressure gases. This will naturally tend to be caused by blow-by, and if bearing tolerances are small, this blow-by loss may be very moderate and worth the bearing work it provides. Additionally or alternatively bearing may by water or other liquids or gases as described earlier by direct supply 1198 as in Figure 217 or via wicks 1199 as shown in various alternative arrangements in Figure 224, supplied by passages 1200. (See Note 2.)

The head may be designed in any manner, including to house conventional poppet valve(s). On reflection it is clear that the central tensile member reduces the possible diameter of the valves, unless four valves are used about a central rod/cable and optionally concentric fuel delivery. A less costly and more efficient arrangement might be the provision of ring valves where a ring valve of median diameter x will

provide double the clearance of poppet valve of diameter x at a given lift. In Figure 226(a) to 232, 226(a) designates an internal head plan view and 226(b) a head cross-section. Figures 226(a) and (b) show a single central ring valve 1201 with twin stems 1202 in guides 1203 provided in bridges 1204 supporting the central portion of the head 1205, in turn supporting the tensile member shown at 1206. The twin stems are linked by means of current split collar construction (not shown) to twin bridges 1207 carrying main stem (not shown) to cam and providing spring 1209 support. Figure 227 shows a similar arrangement but with the valve stem and tensile centers offset by dimensions x and y to more easily permit direct crank/cam valve actuation. The offset may be in one dimension only. Figure 228 shows inner 1210 and outer 1211 ring valves. The outer valve communicates with a charge/exhaust processing volume 1212 separate to that outboard of the head at 1213.

In the "lubrication" of the valve stems the tensile members and the support members 1194 of Figure 224, substances may be used which, when carried to the combustion chamber, affect the combustion process. Fluids which may be used include water, fuel, water/methanol, hydrogen, in either liquid or gaseous state. By "lubrication" is meant the provision of low friction bearing means, including liquid films, gas bearings, etc. Fluid may be delivered to the chamber at the appropriate time by the imparting of pressure to a fluid reservoir (the pressure optionally releasing a valve), causing fuel to leave an orifice communicating with the reservoir. This is essentially the direct injection system employed today, and may also be employed in the new engines. However, in order to eliminate the conventional injector, the head may itself form the injector, housing a conventional plunger 1214 and nozzle 1215, as shown diagrammatically in Figure 229, where the head 1216 is pierced by twin tensile members 1217. Shown alternatively in Figure 230 is a low pressure reservoir 1218 in the head 1216, communicating by optional non-return valve 1219 to the fuel supply line 1220. During fuel delivery a pressure wave is supplied by a plunger (not shown) and transmitted down the supply line opening the non-return valve (which protects the

pump from reverse pressure build up during combustion chamber compression) into the reservoir, causing a fluid jet 1221 or jets 1222 to enter the combustion chamber. In Figure 231 a cam/crank driven plunger 1223 in the cylinder head 1216 either injects fluid 1226 directly into the chamber as at 1224, or indirectly via a very small pre-combustion chamber 1225. If the fluid is combustible and is delivered at above ignition temperature and under pressure, when it comes into contact with the compressed air it will immediately ignite, the resultant expansion causing a jet of burning gas to exit the mouth of the pre-combustion chamber. When either the air in the small chamber is exhausted, or the temperature therein has dropped below ignition due to heat absorption of any latent heat of vaporization of the fluid, combustion in the chamber will cease and the jet of fluid will be delivered through the mouth of the smaller chamber into the larger. An optional depression is shown at 1227, giving increased surface area and therefore heat transfer to the material surrounding the pre-combustion chamber. Figure 232 shows schematically an interior plan view of the cylinder head showing ways in which jet orifices 1228 and/or pre-combustion chambers 1229 may be arranged. This distributed fuel delivery will increase the speed limit at which efficient combustion can be achieved. Figures 233 and 234 show in section and plan section how the plunger mechanism 1230 activating fuel delivery (in turn activated by cam 1231), is greatly enlarged and of kidney shape (to clear tensile member movement). The cam follower 1232 is of a design to permit continuous and variable loading including under high loads. As pressure in the combustion chamber increases, it is transferred to the fluid via the orifices, and from the fluid to a combined camshaft/crankshaft 1233. The loads on the crank during high combustion chamber pressures are in direction 1234, which can be partially offset by the loads transferred to the crank/cam in direction 1235 by the fluid, thus reducing maximum crank bearing loadings. Figure 235 shows a different method of fluid delivery, wherein the tensile member 1236 is used. An annular groove 1237 (or series of depressions 1238) is located in the head 1216, which is filled with fuel delivered under continuous or varying pressure. At a predetermined position a depression 1239 in the continuously moving tensile member communicating with the

combustion chamber will align itself with the depression in the head, causing fuel to flow into the combustion chamber at 1240. As the tensile member moves, fuel supply to the first tensile member depression is cut off, but soon other depressions 1241 and/ or passages 1242 may be aligned with the head fuel supply, providing a controlled supply of fuel to the combustion chamber. The same procedure can be used to supply other fluids (e.g., water) to the combustion chamber. It will be obvious that the pressure of the residual gas in each depression is more or less proportional to the pressure in the combustion chamber at each communication. Depending on fuel supply pressure, the depressions in the tensile member may be wholly or partially filled with fuel, which may lubricate the bearing surfaces between tensile member and head. Instead of an annular groove in the head, a wick 1199 may be provided.

Figures 236 and 237 show in the longitudinal and cross-section respectively an optimized piston 1243 in a twin cup 1244 integral cylinder/head configuration, having a clearance space 1245 shown enlarged in Figure 238. The piston has stiffening flanges 1258. The clearance space is discontinuous, i.e., not annular, although optionally it may be so. Such very small clearance spaces are primarily for variable compression engines. Here a pressure wave during fuel supply causes fuel to be forced through wick 1246 via tensile member depression or passage 1247 into pre-combustion chamber 1248 and thence to clearance space. The two halves of the piston assembly have their joint about the exhaust ports 1249 where combustion chamber pressures re low, and are interlocked as shown to provide accurate location. The arrangement shown has the optional feature of providing for charge purification by residual exhaust gas bleed off. In operation, after the piston has masked the exhaust port, these remaining exhaust gases in the compressing charge being hottest, will rise to the top of the volume and fill the specially provided depressions 1251. As the piston moves up the cylinder, the depressions communicate with the piston void 1252 in turn communicating with the exhaust port. In engine designs where piston blow-by needs to be minimized, special piston grooves can be provided as shown in Figure 239. Cor-

respondingly spaced depressions 1254 are provided in the cylinder wall, which if disposed uppermost will tend to be filled with inert exhaust gas rather than usable charge. It can be seen that as the piston moves up the cylinder to compress the charge, that the pressure in the grooves will always be close to, but a little less than, the charge pressure at that time. Various pressure levels are shown by P1, P2, etc. It is known that the smaller the pressure differential between two gas reservoirs, the slower the rate of gas travel per unit mass between them. Therefore the rate of possible gas travel in piston/cylinder clearance space 1255 (the blow-by) will be reduced.

As an alternative to making the tensile link flexible and so accommodate slack, the slack may be taken up in the bearing(s), so permitting the tensile member to be rigid. If movement in the bearing can be restricted to one dimension, and if the slack-accommodating bearing is such as to always permit transfer of load, then the tensile member may be designed to also function in compression. If the link can transfer both tensile and compressible loads, then two links may share the work of each expansion, reducing the total load carried by a single link, as well by each bearing and by a crank throw at any one time, so permitting lighter construction throughout. In addition a much smoother running engine should result, since the crankshafts are subjected to much more evenly distributed loads. In the example which follows, the crank/tensile link bearing is considered. However any or all of the features described may be equally applied to a bearing between tensile link and the rod of a piston/rod assembly. Figures 246 and 247 show in diagrammatic cross-section two versions of a "stretched circle" bearing permitting take up of slack, where a tensile/compressive link 1282 is integrally attached to non-circular outer bearing shell 1283. Between outer shell and inner bearing 1284 shell is a compressible substance 1285, with Figure 246 showing an intermediate shell 1286 to contain the compressible substance. The intermediate shell may be free to revolve or may be located relative to outer shell by guides, shown schematically at 1287. Any kind of compressible material may be

enclosed at 1285, including elastic ceramic fiber or polymers, springs, etc. In preferred embodiments fluids are used, preferably gases. When a load is applied in direction 1289 the gap between shells at "a" will tend to reduce. If an aperture is provided at 1290 and clearance space at 1291 is minimized, then fluid under pressure will be forced through the gap into main bearing clearance space 1292, providing bearing support. In the case of gas bearings, pressure can be made proportional to load by such means. If in Figure 247 the compressible material is gas and the clearance spaces are kept to a minimum at 1293 then gas pressure on working bearing faces is more or less continuously proportional to load. If it is desired to shift bearing shells rapidly in relationship to each other (range of possible movement shown dotted at 1294) then it is possible to provide a phased pressure relief to provide rapid shell movement. In Figure 247 for example the crank web disc 1295 is provided with apertures 1296 linked by passage 1297 so that as the disc turns in direction 1298 the relative angle to link 1282 to permit both apertures to simultaneously communicate with volume 1288, permitting rapid gas transfer from one side of the volume to the other. As the crank continues to turn the relative angle of 1282 changes to mask one of the apertures and so shut off transfer of gas via the passageway. Figure 248 shows the layout of the variable radii of the interior surface of an outer gas bearing shell, provided with pressurized gas via apertures 1299, so as to permit progressively larger clearance gaps at the perimeter of contact area as the inner bearing shell 1300 approaches the midpoint of its relative movement range. The pressure in the gas bearings may be made directly proportional to the pressure in the combustion chamber (and therefore also partly proportional to the loads on the link) by means of small passages 1301 communicating with the chamber, providing gas access to the highly loaded bearing areas via apertures 1302, either on both sides of the volume (Figure 245) or on one side only (Figure 247). The passage from the combustion chamber may be interrupted by a filter or one-way valve mechanism shown schematically at 1303. A one-way pressure relief valve would permit only high pressure gases to pass in direction 1304, permitting gas bearing pressure to maintain higher than combustion chamber pressure during

portion of the cycle. If it is intended to partly use a gas film between cam and cam follow, one or other can be provided with shrouds to increase pressure build-up between cam and follower as the slope of the cam profile approaches the follower, at the time of peak loads (the follower has to be rapidly accelerated from zero velocity). Figures 249 and 250 show in schematic cross-sections how shrouds 1305 assist in preventing escape of gas at "a" as the cam ramp 1306 turning in direction 1308 approaches the follower 1307, so contributing to gas pressure build-up between the two as contact is made.

In, for example, the case of compound engines, it may be desirable to use exhaust gas at high temperature and pressure to power a turbine, and to have a requirement for exhaust pressures to be low to facilitate two stroke combustion chamber scavenging. In such cases more than one exhaust processing volume may be incorporated in an engine. Figure 251 shows a schematic cross-section of a five-cylinder engine with a high pressure, high temperature exhaust volume at 1308 with exit at 1309, surrounded by a low pressure, low temperature volume at 1310 with twin exits at 1311. Figure 252 shows a schematic layout of a compound system with an IC engine 1312 having ambient air intake 1313, high pressure exhaust 1314 and low pressure exhaust 1315. High pressure exhaust is conducted to a high performance turbine 1316 to exit at 1317 at a pressure approximately matching that of low pressure exhaust 1315 with which it is mixed, and be conducted through low temperature turbine 1318 to emerge at 1319 as close to ambient pressure as possible. Optionally the turbines might be linked by shaft 1320. Figure 253 shows a cross-section of the engine of Figure 251, where high pressure exhaust ports 1321 closable by non-return valves 1322 communicate with high temperature/pressure exhaust reservoir 1323. The piston 1323A when at BDC/TDC unmasks ports 1324 communicating with low temperature/pressure exhaust reservoir 1325. Figures 254 to 256 show a cylinder module made up of three elements, plus piston/rod assembly, valves, etc. incorporating two exhaust processing volumes. The high pressure volume has four shaped snap-in non-return spring loaded valves

1326. The modules are assembled via tensile fasteners 1327 which also attach an evacuated insulating cover 1328, separated from structural elements by trapped air space 1329. Modules are attached to each other via tensile fasteners 1330, with crank cover 1331 attached last at 1332. On the expansion stroke the gases are sufficiently high pressure to open the non-return valves. As the piston exposes the low pressure system the pressure in the chamber drops sufficiently to cause the spring loaded valves 1326 to close. On the compression stroke pressures will be much lower and insufficient to re-open the valves. See also Figures 236 and 237.

It can be seen from Notes 5 and 7 that combustion chamber loads, and consequently bearing loads can be high. If gas bearings are used and gas blow-by is to be minimized, then the bearings may be partially sealed by an oil film. Since gas bearings are generally not operative at low speeds (say less than 500 r.p.m.), this oil film may then serve to lubricate the bearing shells. Of course gas pressure will cause oil loss, but in the basic configuration of Figure 242 this will be burned as fuel. Figures 264 and 265 show schematically a bearing having variable clearance gap 1440, an oil film 1441, an oil-supported bearing area 1442, wicks 1443 communicating with pressurized oil supply 1444 disposed at 90° to direction of maximum loads 1445. Variation of pressure at 1442 would vary support of oil at bearing face.

Note 8 describes some of the stresses which may occur in the cylinder and head elements under high combustion chamber pressures. It is apparent that the stress requirements of the components can be reduced if they are at least partly pre-stressed in compression when the engine is assembled. The forces of expansion will first have to counterbalance those loads before stressing materials to their tensile limits. At present ceramic materials exist which can be used to build engines A and B, with the proviso that when the variable compression/ratio concept is employed with very high maximum ratios, then the technology of today may not be sufficient to provide long-life engines (such as in marine, power generation applications).

There are a number of ways of designing to compensate for peak loads. For example, a fairly strong spring action in the tensile link can act as an energy sink during beginning of expansion, returning work at the low end of expansion. Then, the entire rod/piston assembly can be pre-stressed in compression by a central link (e.g., Figures 216, 220, etc.), increasing net performance under tensile load. If air passages and movement about the pre-tension element is provided, then metal bolts could be contained within high-temperature ceramic piston/rod assemblies.

Generally CAICE units will breath more easily, due to ring valves, unobstructed ports (Figures 254, 255). Turbo lag will be reduced, because of the large crankcase charge reservoir. Although the engines have the widest range of applications, certain embodiments are worth mentioning here. Figure 257 shows a schematic cross-section through a helicopter in normal flight moving in direction 1401. A "ring" engine 1402 is mounted with axis vertical to the ground (enabling any exhaust bleed-off vertically located cylinder depressions to be uniformly positioned in each module). A compressor/rotor/air pump 1403 provides movement of pressurized air down the core of the engine, which partly serves to provide pressurized charge boost at 1408. The remaining air provides vertical thrust at 1404, which is enhanced by the introduction of hot exhaust gases at 1405. The rotors 1406 are attached to revolving ring structure 1407 powered by the engine. A ram scoop is provided at 1409. The central nacelle 1410 is structurally attached to engine and helicopter at 1411 and contains a large parachute which can be fired upward 1412 in emergencies, to clear the rotors. (Such an application of an emergency parachute can be applied to helicopters powered by other engines.) Cross-section Figure 258 shows a "ring" engine 1413 powered torpedo which has a central body containing a warhead 1414 and a reservoir of liquid or other fuel 1415. Water passes through the core of the engine between it and the central tube. Hot exhaust gases exiting in direction 1416 created thrust on their own account and by causing the expansion of water trapped between tube and extension of engine housing at 1417. A similar arrangement can be used for small (say one-man) sub-

marines. IC engine powered propulsion may be of any means including by screws 1418.

The concepts may also serve to provide a new type of hydrofoil craft. They would differ from current practice in that the engine would be under water, probably with the fuel tanks also, so that only air and controls (not drive train) need be provided through the hull supports. Such craft would be much more silent and vibration free than conventional hydrofoils. Figure 259 shows a typical underwater engine pod, attached to hollow hull support 1419 providing air and controls. A "ring" IC engine having three cylinder bands is deployed at 1420, powering both a series of circumferentially mounted propellers 1421 and a ring-shaped turbine impeller 1422. Forward motion 1423 of the engine forces water through the gap 1424 between engine and fuel container 1425. An annular series of charge reservoirs is deployed at 1426, with exhaust gas passing through reservoir 1427, through heat exchangers 1430 located in the water flow through the gap 1424, into nozzles 1428 discharging the gas to produce thrust in direction 1429. Optionally gases can be variably distributed to the nozzles, which may be movable, to provide some steering means. Propulsion is provided by the propellers, impellers, to some degree by the expansion of water in the gap, by the exhaust gas directional jets. Compound engines may be employed to convert a greater amount of total energy to mechanical work on propellers, etc., especially if an independent engine is aboard the hull of the vessel to provide charge boost.

Figures 260 and 261 show schematically a small high speed craft having only a single hull support and single engine pod. Fixed vanes/fins 1431 and movable vanes/fins/rudders 1432 provide stability, maneuverability and lift. The central nacelle 1433 contains fuel, replaceable by water as consumed, so most of the mechanical weight is under water permitting a light and almost completely free hull. In practice the craft would operate like a motorcycle at high speed, being banked around turns by

adjusting the trim tabs (range of movement shown dotted at 1434). In this particular craft engine power is directed to two pairs of contra-rotating screws 1435. Figures 262 and 263 show a larger craft having four retractable engine pods 1436, which when retracted at 1437, are located in shaped depressions in the hull 1438, which still permit the engines to be used when the craft is in the water (as when shown dotted at 1439). The fuel tanks shown may be augmented or replaced by buoyancy tanks. None of the figures are to scale.

Constructions are described in their basic embodiments, without consideration of possible refinements. For example, single chamber multiple fuel delivery points may be activated sequentially to induce controlled turbulence. Water or other liquids may replace the oil in the bearing of Figures 264 and 265. The "stretched circle" bearing may be replaced by a piston/cylinder elastic element in the tensile/compressive link or its bearing.

**4. Efficiency and Performance**

Theoretical efficiency and performance figures have been calculated for single combustion chambers of Engine A and Engine B, identical in every respect, except for degree of charge turbo boost. (See Note 5.) In general these are high speed engines, where ancillary systems are mainly powered via a turbo charger by the exhaust gases, and to a degree by some kind of controlled blow-by of the compressing charge. Oil-less units are assumed. The calculations do not take into account the effects of possible introduction of water, which will tend to produce increased power. Because there is virtually no heat loss, efficiencies should be close to theoretical maxima. (See Note 2.)

The efficiencies per combustion chamber have been extrapolated to Engine 3 (having 3:1 charge boost) in Table 1, which compares new engine perform-

ance with those of present or past engines. In determining bulk, the dimensional guides quoted previously have been used, with a 45% addition for ancillaries. Since all ceramic construction is envisaged, a weight per cubic foot slightly less than that of the conventional aircraft Napier Nomad is assumed. Unlike conventional engines, the new engines will produce more power per firing at speed than when running slowly, since friction losses are negligible but there are proportionately greater gas bearing pumping and blow-by losses at low speed. (See Notes 3 and 4.) To check the validity of the figures for Engine 3, the horsepower per $inch^3$ is compared with that of the Napier. Overall compression is roughly equivalent (boost multiplied by compression ratios of 8.25 x 8 versus 3 x 20), Engine 3 has over twice the fuel/air ratio (1:40 versus 1:19), no oil system and no heat loss, so it should have about 2.25 to 2.5 times the power per $inch^3$ at comparable speeds. In fact the ratios are 1.215 versus 2.48, a factor 2.0412. If the Napier figures are accurate, Engine 3 should perform better than the figures of Table 1 indicate. In making the comparison, weight figures given to the applicant by Volvo were used. As described earlier, the new engines are designed to run at great speed. The speed limit for Engine 3 of 6300 r.p.m. is conservative.

For academic study purposes, the performance of an engine with high speed variable compression ratio, Engine 4, (6:1 charge boost) has been projected. This engine behaves exactly as a fixed compression engine, until the piston takes off after 7200 r.p.m. until maximum speed of 15540 r.p.m., at which its maximum variable compression/ratio of 100:1 (600:1 overall) is reached. (See Note 6.) A separate efficiency and power analysis has been used for engines A and B when operating at that compression/ratio. (See Note 7.) These are the same engines analyzed in Note 5, which operate at those theoretical rates of power and efficiency until piston take off or maximum design compression ratio is reached; as this speed is exceeded and compression/ratios increase so the performance changes toward that at maximum variable compression/ratio, given in Note 7.

The theoretical efficiencies calculated for the 20:1 compression ratio in Note 5 are quite high, but if total energy count is a guide, appear to be realistic. General blow-by and other losses of 5% at 6300 r.p.m., 10% at 2100 r.p.m. and 15% at 1050 have been projected, although they should not be so high. Concerning the calculations for the 100:1 compression/ratios, here the totals are also consistent, but the Ideal Gas Law formulae are no longer reliable. For this reason a 10% reduction has been allowed for, although it should be less. At such pressures there should be very much less disassociation of charge products, with proportionate increase in efficiency. At the high engine speeds envisaged, pumping and blow-by losses should be minimal. The efficiencies are high, but so are the compression/ratios. Where such engines can be used, fuel consumption per unit of power should be exceptionally low.

The calculations represent theoretical conditions. Because processes are not instantaneous as modelled here but continuous, actual maximum temperatures and pressures will be much lower than indicated. Present ceramic materials can be used without difficulty to produce Engines A and B operating at limited geometric compression/ratios of up to 20:1. At present it would be difficult but feasible to build such engines with variable compression/ratios of up to 100:1. Such engines would almost certainly have to have compressive/tensile links, where the work of each compression and expansion would always be shared by both links and both crankshafts, would require some variation of the "stretched circle" gas bearing to assure smooth transfer of loads at BDC/TDC, and would require a considerable proportion of bore area to be occupied by link cross-section. The majority of ceramic materials' strength would deteriorate fairly rapidly with time under extreme temperature and pressure, so long engine life would be difficult to achieve.

As mentioned above, fairly substantial deductions (from 5% to 15%) have been made from the calculated efficiencies to produce projections of possible engine performance. Notes 5 and 7 indicate efficiencies of 58.8% - 58.3% and 75.4% - 75.1%

respectively. Using the formula for maximum theoretical efficiency as a function of compression/ratio (minus the reciprocal of compression/ratio to the power of k-1, plus unity), maximum theoretical efficiencies of 69.3%, 74.6%, 80.2% and 83.5% are obtained for overall compression/ratios of 60:1, 120:1, 300:1, 600:1.

Notes 2, 3 and 4 indicate that at high speeds piston blow-by, gas bearing and heat transfer losses can be very small, less than the 5% - 10% allowed for. This suggests that, with both careful design and operation within fairly narrow parameters, actual uncompounded IC engine efficiencies can be increased to 55% to 75% with proportionate decrease in fuel consumption per unit of power obtained.

With compound engines efficiency should be even greater, since all the available exhaust heat energy is tapped. (In uncompounded engines only that portion of exhaust energy required for charge boost is used.) With turbine efficiencies now in the 50% - 60% range, overall compound engine efficiencies of 65% - 80% are possible. With judicious addition of water and/or turbine after burning, power outputs may be increased still further.

## 5. Applications

The various constructional details described can be combined in different ways to produce engines for a wide variety of applications. For example, where the highest power to bulk or mass is not required, a four-stroke engine with a relatively low speed may be used, which if naturally aspirated may have variable valve lift and timing. Where a lack of vibration is important (e.g., generating engines in research or science environments), a two-stroke engine having "elastic" tensile/compressive crank link may be employed, where work is continually done by each piston on both cranks, providing an exceptionally smooth supply of power. If crankcase size is limited, the "stretched circle" gas crank bearing with compressive/tensile link may be used. With

these designs dimension variation can be accommodated in the bearing, so permitting crank throw diameter to equal or even be less than the stroke. Where high speed engines of fixed compression/ratio are required, then a higher level of turbo charge pressure will speed up combustion processes to match engine speed and will increase permissible engine speed before piston take off. The higher the engine speed (i.e., power to bulk and mass ratios) required, the greater the logic of going to two stroke engines. Again, the smaller the stroke, the higher the engine speed for a given piston velocity and piston take-off point.

If high speeds are required, it would therefore be preferable to have many small combustion chambers than few large ones. Most engines will be direct injected (the high temperatures will tend to cause pre-detonation or knocking in carburetor or indirect injection engines), so will be able to use virtually any fuel.

The application for variable compression/ratio engines are more limited. As can be seen from Note 6, because the reciprocating masses have to be increased substantially to allow for higher pressures, the absolute engine speed limit may not be greatly increased. However, because at a given high speed the variable ratio engine operates under much higher pressures, it will be able also to more rapidly and efficiently combust the fuel. So if for one reason or another the speed and efficiency of combustion needs to increase proportionately with engine speed, then the variable compression/ratio may be employed. The lower the charge pressure, the lower the design compression/ratio, and the heavier the reciprocating parts, the slower the piston take-off speed will be. With unrestrained tensile links compression/ratio will increase approximately to the square of speed, with restraining links the rate of compression rise to speed increase will be much more gradual.

Variable compression engine designs do have special applications. One is for short-life high performance engines, where the engine is only used during the brief

period before the high temperatures and pressures cause the component materials to disintegrate. Such uses would be for motor sport (e.g., drag racing), and missile propulsion, including for torpedoes. Another important application is to use the variable ratio capacity only as emergency stand-by power, with the knowledge that once used for emergency power the engine would be rebuilt. Such uses could be in aircraft (take-off crash avoidance procedure emergency power boost) and in large marine craft. At present such craft are difficult to control in emergency; their stopping distance being less dependent on engine performance than on the function of their speed and mass. However, their provision with a variable compression engine could give them a two- to four-fold emergency additional power supply, permitting them to accelerate, power or maneuver their way out of emergency situations.

The "free piston" variable compression ratio concept is the logical extension of the single piston/twin crank concept, and as such should only be manufactured once fixed compression/ratio twin-crank engines are established. These twin crank engines appear to be most viable in two-stroke form, the latter implying the use of the overhead inlet valve and the mid-piston exhaust ports. For this reason, twin crank/variable link engines can only be properly manufactured when experience in the construction of the single piston/twin combustion chamber engines, preferably with circumferential exhaust processing volume, is established.

Certain of the features described are less appropriate to larger long-life engines, and more suited to smaller or shorter life units. Such units would include those used for mopeds, chain saws, highway sign power generation, standby emergency power, outboard or inboard small marine craft. Here the use of tensile yarn, etc. is feasible.

The variable valve actuation capability has many useful applications, apart from assigning volumetric efficiency in wide speed range naturally aspirated engines. In

two-stroke engines, variation of inlet valve actuation may be used to compensate for the reduced charge-to-exhaust-pressure differential required at lower speeds. In all middle to high compression/ratio engines, inlet valve variation may be used to lower effective compression/ratios during cold start or idle. In engines where there would otherwise be too much energy remaining in the exhaust gases, the variation of inlet actuation (or indeed the fixed adjustment of inlet opening) may be used to cause some of the charge to be bled back to an intake gas reservoir, so reducing effective compression/ratios, but maintaining expansion compression/ratios. This technique is sometimes referred to as "overlong" expansion, and serves to direct a greater percentage of total fuel energy to work on the piston.

In the case of true compound engines (such as for aircraft), it may be desirable to have hotter exhaust gases at higher pressures. There the use of the variable compression concept seems inappropriate; compression/ratios should be reduced as far as practical. In the case of two-stroke engines, a two-stage exhaust system is advisable if high charge boost pressures are not used.

Hopefully the foregoing has established that the various features described can be combined in any way to produce a complete new generation of more efficient internal combustion and compound engines.

The cost of the engines described should range from approximately equal to about three times that of conventional engines, on a cost per pound or per cubic foot basis. On the one hand the engines have fewer and often simpler components, on the other expensive machining of ceramic materials to close tolerances will usually be required. However the new engines will have so much greater power to bulk and power to weight ratios, that the costs per unit of power should be many times less than that of conventional units. As can be gathered from Table 1, in the case of large generating or marine propulsion engines the cost reduction could be very substantial.

Potentially important advantages of the new CAICE units concern packaging. As pointed out previously, the engines should vibrate less than conventional units. They should also be much more silent, due to the heavy insulation with which they are surrounded, and to the fact that the principal sound generator - the exhaust system, is now in the interior of the engine. As can be seen from Figures 189 to 200, the units can be rectangular, and because no air circulation is required, can be installed in situations not previously feasible. For example, in automobiles and light trucks, they could be installed under seats, or within double skin floors.

## AN IMPROVED TRANSMISSION SYSTEM

A previous section discloses engines which may have two crankshafts with possible high power outlets per shaft. Of course these may be linked and power taken from one shaft. However, in such cases it would be preferable to couple such engines to transmission system which can be designed to have two input shafts, as the transmission discloses herein.

The present invention comprises a transmission system capable of providing a continuous flow of power at ratios infinitely variable between fixed parameters.

An object of the invention is to provide a particular transmission suitable for inclusion in the majority of motor vehicles. The majority of previously known variable transmission devices are generally limited to the amount of power which is transmissible and are therefore not suitable for vehicles. The present system comprises a direct mechanical variable transmission suitable for including in, for example, highway trucks. In addition to providing a variable transmission function, the various elements of the present invention may be arranged in one unit to additionally allow that unit to

act as a clutch, a reversing mechanism, a differential, a power take-off source, a variable load distributor (say varying load to front and rear wheels in a multiple drive vehicle), all these functions performable if desired by the single unit and where appropriate simultaneously.

The invention comprises a series of rollers of controlledly variable diameter connected by a flexible friction member, such as a band, each of the rollers communicating with input, intermediate and output shafts as desired. The rollers in the majority of embodiments will have a constant diameter at any point of length. but that this diameter will be variable. It will be apparent that by such a system a variable mechanical transmission is achieved capable of transmitting high loads with low losses. The reasons are that a band of inter-connecting rollers will have very large areas of contact (i.e., load transmission), compared with say band and cone variable transmission systems, and that this contact area will not be such as to cause differential slippage, as say in wheel and disc systems of drive.

The basic principles are illustrated diagrammatically in Figure 266, wherein in solid line are shown roller A of diameter unit 2 driven by power input shaft. the roller communicating by endless belt C (tensioning not shown) with roller B of diameter 3 units which drives the output shaft, resulting the latter operating at ⅔ the speed of the input shaft. If roller A is increased to diameter 3 units and roller B is simultaneously reduced to diameter 2 units, as in the arrangement shown lightly dotted. then the output shaft will be turning at 1½ the speed of the input shaft, 2¼ times as fast as in the earlier arrangement provided input shaft speed has remained constant. If the respective increase and reduction in roller diameter had been to 2 units and 4 units, then the output shaft would have increased its speed four times. In operation it is intended that such variation in gearing takes place during power transmission, and that the gearing ratios be infinitely variable between two extremes.

June 24 '90

The following diagrams accompany the text by way of example, wherein:

deleted

Figure 266 illustrates the stepless variable ratio principle,

Figures 267 to 275 illustrate various embodiments of transmission system,

Figures 276 to 277 illustrate an embodiment of variable diameter roller,

Figure 278 illustrates relationship between two rollers,

Figures 279 to 289 illustrate details of the first roller embodiment,

Figures 290 to 293 illustrate the principles of a second embodiment of variable diameter roller.

The roller may be arranged in any way or combination of ways to form the transmission system of the invention. Various arrangements are shown by way of example. In Figure 267 is shown in diagrammatic cross-section a transmission system which may act as a clutch, having two expanding rollers 1, 2 connected by an endless band 3 longer than needed to make the drive between rollers. Idler rollers 3 acting as belt-tensioning members are provided to move in direction 5. It is apparent that if input roller 1 is being driven and idler rollers 4 are in the withdrawn position causing the belt 3 to be slack, then output roller will not be driven. By gradual tensioning of the belt by means of moving the rollers 4 inward in direction 5, the drive will progressively be taken up, thereby causing the system to act both as clutch and variable drive.

In Figure 268 there is shown in diagrammatic cross-section a transmission system capable of acting as a differential, wherein input roller 1 is connected by means of endless belt 3 to two output rollers 6, 7, the latter's capacity for expansion and contraction being counterbalanced by mechanical linkage 8. For example, if both rollers are spring-loaded to bend to increase their diameter, and if the expansion and contraction actuating mechanism of rollers 6 and 7 are linked, then if roller 6 is caused to contract roller 7 will automatically be caused to expand. If the assembly is filled to a vehicle and roller 6 is connected to the left wheels and roller 7 to the right wheels.

then the system can be adapted to function as vehicle differential and variable transmission combined. As will be seen later, the rollers of the invention may in some embodiments be spring-loaded to expand against belt tension. An increase in belt tension and therefore loading will cause the rollers to contract. Such rollers where variation in load will cause variation in diameter may be used as the output rollers 6 and 7. Alternatively, variation in load may directly or indirectly actuate the roller to expand and contract.

In Figure 269 there is shown in diagrammatic cross-section an arrangement whereby power distribution may be varied. A power input variable-diameter roller 1 is connected by means of endless band 3 to two output rollers 6, 7, say respectively driving the front wheels and rear wheels of a four-wheel-drive vehicle. There are two tensioning rollers 10, 11, capable of movement in direction 5. As shown rollers 10, 11 are so positioned as to cause a greater proportion of the total contact between band and both output rollers to be between band and roller 6, resulting in a greater quantity of total power to be transmitted to output roller 6 and a lesser quantity to output roller 7. By movement of tensioning rollers 10, 11 to the position shown dotted roller 7 may receive a greater proportion of total power than roller 6. By these or other means the degree of band "wrap round" a roller can be varied, so varying the quantity of power transmitted. The assembly could be used to say provide more power to the rear wheels during acceleration and/or to take more power from the front wheels during braking.

Figure 270 shows in diagrammatic cross-section an assembly functioning as a variable ratio transmission and a three separate differentials, suited say for a four-wheel drive cross country vehicle. Input roller 1 driving by endless belt 3 two pairs of rollers, D, E wherein the upper in each pair, 12, 13 drive the rear wheels and the lower 14, 15 the front wheels. Pair D drives the left side of the vehicle and E the right, the pairs being linked by means of mechanism 16 in the manner of Figure 268 to

form a differential between left and right sides of the vehicle. The ends of mechanism 16 communicate with secondary mechanisms 17, 18 linking the rollers of each pair in the manner of Figure 3 so that a differential between front and rear wheels is formed. By the principles shown by way of example in Figures 267 to 270, a single assembly having the rollers of the invention may act as a clutch, a variable ratio transmission a means of providing multiple separate drives differentiated from one another, and also provide a means of variably distributing power between any combination of or individual drives. In the above description and elsewhere in this disclosure unless stated to the contrary, the input and output rollers are variable diameter rollers. However, the principles of the invention work equally if one of two rollers in any system or subsystem is of non-variable diameter.

If the output system consists of a multiplicity of rollers, it may be desirable to have more than one input roller, so that the contact areas of input and output systems become more equivalent. Figure 271 shows in diagrammatic-cross-section a system having four output rollers 17 connected by endless belt 3 moving in direction 18 driven by two input rollers 19 driven from a central shaft and gear wheel shown dotted 20 turning counter-clockwise, by means of gears shown dotted 21 mounted fixedly and concentrically on input roller shifts, these last turning in a clockwise direction. Figures 272 and 273 show how it is possible to compensate for naturally reducing contact area due to reduction of shaft diameter. Figure 272 shows roller 1 reduced and roller 2 expanded, connected by endless belt 3 tensioned by movable idler rollers 22, positioned close to roller 1 so as to cause a "wrap around" effect and so increase contact area of roller 1. As gear ratios change and roller 1 diameter expands with a balanced reduction of roller 2 diameter, the idler rollers 22 move to the position indicated in Figure 273, arrows 23 indicating range of movement of the idler rollers, and not any tensioning force.

In some transmission systems it may be desirable to incorporate reversing means, two of which are illustrated by example in Figures 8 and 9. In Figure 274 input roller 1 turning clockwise drives output rollers by means of intermediate movable rollers 24 turning counter-clockwise and endless belt 3 tensioned by idler roller 25 capable of movement in direction 26. In the arrangement shown in solid like the band and rollers 2 will rotate in a clockwise direction, but when roller 24 are moved in direction 27 to positions 28, the band will make a direct contact with the input roller 1 causing it and rollers 2 to be driven in a counter-clockwise direction. Figure 275 shows two input rollers 29, 30 turning in counter directions, adjacently mounted on a pivotal carriage indicated diagrammatically by line 31, the carriage being so positioned as to cause roller 29 to make contact with an endless belt 3, part of a transmission system of the invention. When the carriage is pivoted through direction 32 to a new position shown dotted roller, 30 is caused to make contact with 'the belt, so causing it to move in the opposite direction.

It is proposed to disclose below at least two alternative embodiments of a roller having substantially continuous surface and variable diameter. In the first configuration there are two cones slidably and engagably mounted on a shaft through their axis, sharp ends facing one another. The cones have projections or depressions formed running between the narrow and wide ends a series of members having each end slidably mounted on corresponding depressions/projections of each cone. In operation the cones actuated to move toward one another, causing the members to move away from the axis of the shaft. The members are of such configuration that they form the effective surface of the roller assembly, so by the above movement the diameter of the roller is increased. Similarly, by causing the cones to be moved apart the diameter is reduced. These principles are illustrated diagrammatically in Figures 276 and 277, showing in the former case a roller assembly with diameter reduced and in the latter case with diameter enlarged. Cones 50 and 51 are only slidably mounted on shaft 52--they are not free to rotate relative to the shaft--and have on their surface

depressions shown simplified at 53. Spanning between corresponding depressions on the cones are a series of members 54 which support the drive belt shown in outline at 55. The assembly is shown with cones moved toward one another, to expose key means 56 mounted on shaft 52, and to cause the members 54 to be moved radially outward from shaft axis 57. In both the basic embodiments of the roller of the invention, work must be expended to cause the roller to expand against the likely load of endless band under tension. It is for this reason that rollers are best arranged in pairs so that one expands when the other contracts and vice versa. Figure 278 shows diagrammatically by way of example how this can be regulated, especially in the case of differential type mechanisms. A pivot 60 with rounded thrust ends 61 bearing on collars 62 of one cone of each shaft causing the loads to be balanced between rollers is shown in a position wherein roller 1 is enlarged and roller 2 reduced. When the sizes of the diameters of the rollers is reversed, the pivotal link is shown in a new position, all shown dotted. The pivotal points 63 are connected by a structural mechanical element indicated by arrow 64, which in simple embodiments will be a rigid member. In complex embodiments, as for example in that of Figure 270, this element may be somehow of variable dimension, thereby causing the mean diameter of the brush roller assembly of Figure 11 to be variable.

The members spanning between the cones may be of any convenient shape or form, but in preferred embodiments re "T" or "I" or "L" shaped in cross-section, and have extremities capable of overlapping one another, in a manner which can be described as a kind of reversed iris-type action. This is illustrated in diagrammatic cross-section Figure 279, where members or segments 70 making up the roller are shown arranged in the reduced diameter configuration in solid line and in enlarged diameter configuration in dotted line 71. Figure 280 illustrates in cross-section an embodiment of a load bearing cone 72 supporting one end of each segment member (not shown), where the cone is keyed to input/output shaft 73. The cone has a series of grooves 74 formed radially/axially along its conical surface, which in operation

receive rod- or nodule-like projections which form part of or support the segment member, shown in diagrammatic elevation in Figure 281, the left half representing a member having high mounted bearing projection 75, and the right half a low mounted bearing projection 76. In this case the segment is supporting a segmented endless belt, wherein the different segments 77 of the belt may move relative to one another, being linked by floating bridging pieces 78. Figure 282 shows diagrammatically a conical end piece which is not a true cone, consisting of a collar 79, slidably and keyedly mounted to a shaft 80, on which are mounted a series of radial/axial fin-like projections 81 of roughly triangular shape, being joined to and supported by one another by means of a stiffening web 82. The extremity 83 of this fin forms a bearing for the grooved support end 85, of the segment member 84, the assembly in operation causing the member 84 to slide along the fin extremity 83 to say the position shown dotted.

Figure 283 shows by way of example a cross-section through two segment members 90 as they might be positioned in an expanding roller assembly relative to its axis 91 and each other. The members 90 are basically "L"-shaped, having a more or less radial or perpendicular portion 92 designed to transmit substantially tensile or pressure 93 loads and another more or less arcuate portion 94 designed to transmit substantially thrust loads 95. These loads are of course transmitted by or through the band, shown dotted at 99. A bearing nodule of the member is shown elevationally at 96. The members may be linked to each other by male/female guides as at 97 or by tension and/or compression springs as at 98. Additionally or alternatively the members may be connected to the cones and/or shaft by any manner of springing or loading means. Figures 284 to 286 show diagrammatically how the relationship between arcuate portions of the members varies between reduced roller diameter operation at 284, intermediate at 285 and enlarged diameter operation at 286, where each extreme member tip 100 is shown resting on a "catching" lip 101 adjacent to the elbow 102 of each member. It is of course the arcuate portion of the member which will form contact with the band, and to this purpose such portion of the member may be of any

convenient form, material or composition. By way of example there is shown in Figure 287 a segment member having a composite construction and dimpled surface. In heavily-loaded applications it might be desirable to incorporate ball or roller elements 103 at member tip 100 and/or elbow 102, or at some intermediate points. The segment members have been shown overlapping, but in other embodiments of the invention they may be non-overlapping to give a roller which has always or sometimes a substantially discontinuous or slatted surface. An advantage of the slatted roller is that the contact area between roller surface and band can be kept relatively constant despite variation of diameter. Figure 288 illustrates in diagrammatic cross-section a segment member 104 suitable for a slatted roller application. The cross-sectional forms of the segment members may be of elements combined to form "T" or "I" or "L" shapes. The junctions of these elements have been described as being rigid, but in alternative embodiments the junction or other points of the cross-sectional form may be hinged or designed to have greater or special flex. For example such hinge means may be incorporated at elbow 102 in Figure 286 or at 105 in Figure 289.

The alternative embodiment of variable diameter roller also has segment members--which may be of overlapping or slatted form--but they are not supported at the ends by conical forms. Instead they are supported at optional points on their length by a corresponding series of linkage systems carried at at least two points on a rotating shaft, the position of the segment member relative to shaft axis being determined by variation of the distances between two or more points of contact between shaft and each linkage system. The principle of operation is shown in diagrammatic cross-section Figure 290, only one of each of two alternate segment member types being shown for simplicity of illustration. A shaft assembly 110, to be later described in detail, supports each segment member by two differing linkage systems (shown above and below the shaft respectively), each system being supported on the shaft at two pivotal points 111, 112, communicating with major lever 114 and minor lever 115 respectively. The shaft assembly has the special feature that the distance 113 between

points 111 and 112 is variable. The minor and major levers intersect at pivot 116, the major lever extending to carry the segment member 117 either rigidly mounted as at 118 or pivotally mounted about 119. It can be seen that variation of distance 113 will cause the ends of major levers to move further from or close to shaft center 120, thereby causing the variation of the diameter of a roller composed of a multiplicity of preferably uniform segment members and associated linkage systems. Any useful design of shaft assembly having mounting points of circumferentially varying distance may be employed in the invention, but in a preferred embodiment a shaft having thee concentric elements is used, as shown in diagrammatic cross-section Figure 291 and elevation Figure 292. The center 121 of the three concentric elements is the main portion of the shaft carrying the input/output loads, having disposed within a slidably mounted shaft 122 prevented from rotating independently of main shaft 121 by means of key 123 projecting through elongated axial slot 124 in main shaft, the key projecting beyond main shaft through a diagonal cross-axial slot 125 in the outer shaft collar 126, the latter being by some mechanical means restrained from axial movement relative to main shaft, only being rotational about main shaft. The collar has other slots 127 through which main shaft pivotal lever mountings 128 project, the collar having attached to it separate pivotal lever mountings 129. It can be seen that by axial movement 130 of shaft 122 relative to main shaft 121, the circumferential distance between mountings 128 and 129 can be varied. Because a variable diameter roller is likely to have a multiplicity of segment members and corresponding linkage systems, it is preferable to arrange for the linkage systems 131 for each segment member 132 to be axially staggered, as shown in diagrammatic elevation Figure 293.

The features and characteristics described above may be used in any combination to carry out the invention. The components of the transmission assembly may be of any convenient construction or material, including of metal, plastics or ceramic material. The latter is considered particularly suitable for some embodiments because of its low weight in relation to compressible strength. Tribology may be by

any convenient means, including by dry or wet systems, or by a combination of both, or by a dry band connecting wet-lubricated rollers and actuating gear.

## VARIABLE CAM SYSTEMS

As mentioned in other sections, in many of the engine embodiments described, it would be advantageous to provide variable timing life and dwell of valve actuating mechanisms, of fuel delivery mechanisms, of spark or other ignition mechanisms. In addition, such variable valve actuation can be used to provide charge bleed-off during compression stroke to realize variable effective compression/ratio engines, to improve two-stroke charge purity, and to provide "over-long" expansion ratios. as described elsewhere.

The invention comprises the novel arrangement of a number of separate features, some of which are known, to give an engine which has a variable effective compression/ratio and/or variable effective capacity. An alternative embodiment of the invention provides an engine having working chamber valves of variable lift and opening duration.

The desirability of having engines of variable capacity, cam/valve gear and compression/ratio has long been known and is evidenced by the number of patent applications taken out on these subjects. It is proposed here only to summarize briefly the known advantages such concepts entail.

A controllably variable compression/ratio engine would enable ratios to be varied when the engine is cold, or being fired, thereby making starter easier. The manner and degree of variation desirable depends greatly on the type of engine employed. For example, in the case of say a super- or turbo-charged compression-

ignition engine, an increase in starting pressures would compensate for lack of charge pressure due to the charger being inoperative. A further, perhaps more important advantage of a variable compression/ratio engine would be to burn fuel at what was always the optimum ratio for a given operating condition, so saving fuel. (Generally, the higher the compression/ratio, the greater the amount of work derivable from a given amount of fuel.) Engines are constructed to the compression/ratio at which they will properly function under the most disadvantageous condition, usually low revolution with high load. In fact this condition usually occurs for a small proportion of total operational life, so by varying the compression/ratio upwards during other operating modes, more work is derivable from the same fuel. To the knowledge of the applicant. prior designs for varying engine compression/ratios involve physical rearrangement of engine geometry, variation mostly not being easily and instantly achievable while the engine is running. The present invention involves no alteration to engine geometry, and is therefore a major improvement over costly, bulky and impractical known designs.

The principles of the invention may be so considered and embodied as to provide a variable effective capacity engine. Such optional and controlled variation of capacity has self-evident advantages where fuel economy is important. Since engines are today built to a capacity to provide the maximum power likely to be required, an ability to lower effective capacity when maximum power was not required could lead to major fuel economy and increased engine life (due to combustion loads being smaller).

Currently the vast majority of engines have valves at fixed settings which are a compromise of the various ideal settings for various operating modes. It is therefore evident that the provision of a controlledly variable camshaft would enable valve setting to be at optimum under all conditions, thereby allowing for improved volumetric efficiency and consequently better fuel economy and/or easier compliance with exhaust emissions standard under various operating conditions.

A further object of the invention is to provide a means of improving charge composition control in two-stroke engines, as will be described subsequently. The efficiency of two-stroke engines depends to a considerable degree on the proportion of air/fuel mixture and exhaust gases in the charge, and how the relevant constituents are mixed and distributed during combustion. An improvement in charge composition control could therefore lead to improvements in mechanical performance and consequent fuel saving.

As has been mentioned, there is a considerable prior art in the field to which the invention relates, which has resulted in various separate features becoming known - as for example the cam having a lobe of conical axial profile. The applicant makes no claim for the known separate principles and components of the art, but solely for the way these disparate elements have been adapted, expanded, augmented and integrated to constitute a novel engine, which the applicant feels is the first really practical engine configuration embodying variable compression/ratio capacity or valve gear principles. Because of the increasing standards of mechanical and thermodynamic efficiency now demanded, it is felt that the invention will have wide application.

The objects of the invention described above are, considered to include by implication the provision of an engine capable of using differing fuels.

The invention relates to a variable effective compression/ratio engine, a variable effective capacity engine, an improved two-stroke engine and a means comprising cam and follower arrangement for causing valve timing and lift for the above as well as any other engines to be optionally and controlledly varied, including while the engine is operating. In this specification, by engine is meant any form of engine employing an expansion cycle including pumps, and by valve is meant any type of controlledly enlargeable and reducible aperture or passage, including for example poppet, ball, sleeve and butterfly valves, slides, gates, etc. The invention most specifi-

cally relates to combustion engines of the internal type, including two-stroke and four-stroke; reciprocating and rotary; diesel and gasoline; supercharged, turbocharged and naturally aspirated; carbureted and fuel injection engines. The invention also relates to external combustion engines, such as the steam. Stirling and rankine-cycle types.

The invention comprises an engine including an arrangement for varying its effective compression/ratio and/or effective capacity by means of optionally and under control bleeding off some of the induction fluid already aspirated, or by means of restricting or assisting aspiration of the induction fluid, the said arrangement comprising at least a cam capable of axial and rotational movement, the said cam having one or more lobes of progressively varying cross-section. Described below are various embodiments of the invention, in which distinction will be drawn between the variation of compression/ratio and capacity, and the provision of a valve mechanism providing improved control and/or variation of working fluid entry and exit to the working chamber. This latter control and/or variation, effected by means of variable valve timing and opening duration, can be said to a degree to provide an effective variation of compression/ratio and/or capacity. According to the invention, the effective compression/ratio and/or capacity is varied by the following means. An engine is constructed so that its parts operate to a fixed geometry, causing a given amount of charge to be aspirated, this quantity being called the swept volume. In conventional parlance swept volume refers to the geometrical configuration of the engine, being defined as bore multiplied by stroke, further defined as capacity, but in this specification it will refer to the quantity of charge actually aspirated under a normal or designated standard or optimum working condition, being further described herein as effective capacity. Thus an engine having a bore of cross-sectional area of 50 square centimeters and stroke of 10 centimeters will have a conventionally-described swept volume and capacity of 500 cubic centimeters, although in fact if it is a naturally aspirated engine it will contain less volume of charge measured at externally ambient density, say only 450 cc, and if it is a force aspirated engine it will have more charge,

say 800 cc, these actual charges being referred to herein as the swept volume or effective capacity. If the engine described above has a working volume at top dead center of compression stroke of 50 cc, then that engine will have what is understood as a compression/ratio of 10 to 1. But this refers to the geometric arrangement of the engine, and it can be seen that the two examples referred to will have a real or effective compression/ratio of 9 to 1 and 16 to 1 respectively.

According to the invention, it is proposed optionally and controlledly to vary this effective compression/ratio and/or compression/ratio by the bleeding off of some of the aspirated charge during the compression stroke. It can be seen that if 50 cc were bled off in the case of the naturally aspirated engine sited above, then only 400 cc of charge would remain to provide work, giving a reduction of effective compression/ratio from 9 to 1 down to 8 to 1, and a reduction of effective capacity from 450 cc to 400 cc. It is proposed that this bleeding off is affected by the optional and controlled opening of a purpose-provided valve communicating with the working volume, or in a preferred embodiment by the second opening of a valve already provided in the engine, such as the inlet or exhaust valve, means for achieving such secondary opening being described later. Alternatively, the bleeding off may be affected by the optional and controlled delayed closing of the inlet valve until substantially during the compression stroke. If the bleeding off is achieved by the secondary opening of the inlet or exhaust valve, the timing is preferably arranged to coincide with the "pulse effect," that is with the partial vacuum that will have been formed behind the valve either by the retreating pressure wave caused by the charge rebounding against the closed inlet valve or by the momentum of the exhaust charge.

An alternative method of varying the effective compression/ratio and capacity of an engine is by optionally and controlledly hastening the closing of the inlet valve, or reducing the degree of opening of the inlet valve, thereby reducing the charge

aspirated. This will cause a throttling side effect, and in a majority of engine types will be less mechanically efficient than the means described earlier.

The variations may be effected by automatic or manual means; may be dependent on such factors as engine speed, load, the desire to prevent the formation of exhaust pollutants, ambient temperature, atmospheric pressure, quality of available fuel, warm starting, cold starting, the desire to reduce engine power under certain conditions without proportionate reduction in engine speed or any other factors. The variation may be made between operations of the engine or during operation, and may be continuous and infinite between two designed polarities. In a majority of cases the engine will be designed to operate at one polarity with progressive variation to another polarity. In other words it may be built to operate normally at maximum compression/ ratio and capacity with reduction when certain running conditions make it desirable, as in the case of say a motor vehicle naturally aspirated engine, or the engine may be built to operate only at maximum ratio/capacity under special conditions such as starting, there being a reduction under normal running, as in the case of say a forced charge marine engine.

The means of carrying the invention into effect comprise at least a cam capable of moving past its follower in two directions, rotationally and axially, either separately or in combination. The cam may have one, two or more operational lobes, either separate or merging with each other. The follower to be used with such cam may be either rounded or dome shaped and free to rotate on its axis, or it may be fixed say by keying means to provide a cross-section, taken on a line parallel to the cam section (whether flat or curved in the other direction), which consists of two or more distinct planes, whether these planes are themselves curved, shaped or flat.

Figure 294 shows, by way of example, a cam 1 mounted on a slidable camshaft 3a, the primary lift profile or lobe 4 of progressively varying extent being

shown elevationally. It can be seen that the variation of longitudinal position of the cam relative to the fixed roller will result in the rotatable dome-head cam follower 2 being lifted to varying extent. The cam 1 has a much smaller secondary profile or lobe 3 of progressively varying extent, shown elevationally in Figure 295. Sections A to D in Figure 294 are shown respectively in Figures 296 to 299, illustrating the progressively varying nature of both profiles. Figure 295 shows an alternative follower 5 which is not free to rotate, and which has three separate planes, a secondary lobe contact plane 6, a clearance plane 7 and a primary lobe contact plane 8. Such type of follower is shown in greater detail in Figure 332. The cam and follower arrangement shown in Figures 294 through 299 would be suitable for mounting in a four-stroke automotive engine in order to provide a secondary lift to say the inlet valves to bleed off some of the aspirated charge, where the relationship of section Figure 296 corresponds to high revolutions with maximum effective compression/ratio and capacity and that of section Figure 299 to low revolutions at high load, giving minimum effective compression/ratio and capacity, this variation of ratio/capacity being determined by whether and to what degree the secondary lift is operative. Because the cam moves in two dimensions, the opportunity has been taken to make the main inlet lobe variable too, to provide the inlet valve with a lobe having extent proportionate to say engine speed and load. The secondary lift features may be eliminated to give an embodiment having one variable lift per cycle to the inlet and/or exhaust valve of an engine. This configuration is illustrated diagrammatically in Figure 301, which is a perspective view of a am having only one lobe varying in lift and/or dwell. Such a cam may be used also to control the optional late or early closing of a valve to bleed off from or restrict entry of charge to engine working volume. Figure 303 shows a similar cam, but having a lobe of longitudinally twisted or axially inclined profile, to provide a variable valve timing. The feature illustrated in Figure 303 may be incorporated with any embodiment of the invention. Figure 300 is a perspective view of a cam suitable for the actuation of a special valve to bleed charge from the working chamber of an engine, in this embodiment having a lobe progressively reducing toward a portion of the cam where there is

no lobe or lift. This feature may also be incorporated with any embodiment of the invention. Figure 302 shows a cam having multiple lobes, progressively increasing in parallel, instead of in opposition as in the embodiment of Figure 1.

Figure 304 shows, by way of example, an arrangement for laterally moving a camshaft in accordance with engine speeds and load. The camshaft 3a is mounted in bearing 6 and seal 7 assemblies. One end of the camshaft 3a is connected to a pair of centrifugal weights 8 (here shown in simplified and diagrammatic form) which tend to pull the camshaft to the right as its rotational speed increases owing to the centrifugal force developed. The other end of the camshaft is connected to a piston 9 slidable in a closed-ended cylinder 10. Optionally a compression spring 11 acts against the centrifugal force rightward pull on the shaft, to be counterbalanced by compression spring 13 communicating with setting adjusting screw 14 and lock-nut 15. An outlet 12 is connected to the inlet manifold, an increase in engine load by means of manifold depression in chamber 10 causing piston 9 and attached camshaft to move to the left. Thus, if the cam of Figures 1 to 3 were attached to the camshaft of Figure 304, variation in valve timing and lift, effective compression/ratio and capacity, could be automatically governed by such factors as engine speed and/or load. For simplicity's sake, drive to the camshaft has not been shown. This could be via splined and toothed collar 16 to chain 17 as shown diagrammatically in Figure 305, or by gears having teeth of greater than normal extent, as shown in Figure 306. Alternatively, as shown by example diagrammatically in plan in Figure 307 and elevation Figure 308, the camshaft may be driven by angled chain 17, here via sprocket 18 fixedly attached to camshaft 3a via angled intermediate sprocket 19 from drive sprocket 20. In another embodiment illustrated in Figure 309 the chain is not angled, but of such link design and length as to permit lateral movement. The embodiment of Figure 304 has shown the lateral movement of the camshaft governed by engine speed and load, but of course it may be automatically governed by any other parameters or combination of parameters, including for example engine temperature, exhaust emission control considerations, etc.

The foregoing disclosure has described the basic principles of the invention. Before describing certain constructional details mostly relating to cam followers, which in no way add to these basics, it is proposed to describe by way of example various applications and embodiments of the invention.

The secondary opening of exhaust or inlet valve or opening of another say special valve, during engine stroke or in some embodiments the expansion stroke, has many useful applications. In the case of a secondary opening of the exhaust valve, the bleeding off of some of the charge may be used to better effect exhaust emission control, especially if thermal or catalytic reactors are employed in the exhaust. This technique is preferably employed in direct ignition engines, where the charge need not contain fuel at the time of bleeding off, to provide extra air or oxygen to the exhaust system. In some embodiments the bleeding off of a charge containing fuel may be employed to create combustion, and consequently extra heat, in an exhaust reactor, where the reactor is of suitably robust construction and the combustible charge and residual exhaust gases are suitably balanced and mixed to reduce the risk of detonation. In many possible applications an engine of the invention will be so constructed as to bleed off part of the charge during certain conditions, such as low speed/high load, where an increase in combustion mixture is likely to occur. If the bleeding of is via the exhaust valve, then a supply of air will simultaneously be supplied to the exhaust system to assist in the reduction of partly combusted constituents associated with rich mixture. However, in other embodiments or situations the invention consists of bleeding off charge to the exhaust system with the sole object of regulating exhaust emissions, that is not necessarily to affect engine performance directly.

A further embodiment of the invention consists in arranging for the bleeding off or restriction of charge to an engine in order to provide a multi-fuel engine. It is known that it is desirable to have such engines capable of combusting a selection of available fuels, especially in military, overland or exploration vehicles.

According to the invention, an engine is constructed to combust for example diesel fuel at highest compression/ratio without bleeding off, bleeding off optionally and controlledly arranged when fuels of higher viscosity are used. For example, an engine may be constructed to operate using diesel fuel at an effective compression/ratio of 16 to 1, which is reduced by bleeding off of a charge to 10 to 1 when gasoline is used.

In another embodiment of the invention, the process of bleeding off part of the charge, either by secondary opening of exhaust or inlet valve or by the opening of another valve or by the delayed closing of the inlet valve, is used to improve control of charge composition. For example, it is known that in internal combustion engines the charge consists of a combination of fresh fuel/air mixture and residual exhaust gases from the previous combustion. In four-stroke engines proper design of valve timing and opening can reduce residual exhaust gases to a quantity where they do not significantly affect engine efficiency, but this is not the case with two-stroke engines where the charge always contains a significant amount of residual exhaust gases. Since there is nearly always a controlled or designed fluid flow through the combustion volume, the proportion of exhaust gas present when the valves are closed will tend to be greater in a particular locality, usually near the exhaust valve. This will be especially true in engines where the exhaust valve is at the top of the combustion chamber, where the exhaust gases previously more mixed with charge will have risen during the compression stroke by virtue of their high temperature compared with incoming charge. to form a layer adjacent to the exhaust valve. The invention consists in giving the exhaust valve a secondary opening during the compression stroke to cause most of the layer or pocket of the residual exhaust gas to be expelled. thereby improving the purity of the charge at the moment of combustion. It is realized that the volumetric efficiency of the engine is thereby moderately reduced, but this can be compensated for by increasing the quantity of fresh charge aspirated, especially in the case of two-stroke engines where the charge is nearly always force induced to some degree and adjustments are relatively easily made. By way of example there is shown in Figure 310 in

diagrammatic cross-section a two-stroke combustion chamber 21 having lower inlet ports 22 and overhead exhaust valve 23, the piston 24 shown about halfway up the compression stroke. The valve 23 has a special domed head to allow the hot residual gases shown shaded at 25 to gather at the highest point 26 via the chamber at the perimeter of the valve, which is given a brief secondary opening during the compression stroke. Figure 311 shows in diagrammatic cross-section a two-stroke combustion chamber 29 having cross-flow from inlet ports 27 to exhaust ports 28, the piston 30 being at bottom dead center. There is provided an additional exhaust port 31 closable by say slidable gate 32 communicating with exhaust gallery system 33. If bleeding off of residual exhaust gases (which will tend to collect above main exhaust port) is desired during the compression stroke the gate 32 may be left wholly or partly open during engine operation, or may wholly or partly open and close during each stroke. The gate may be operated by any means, manually or automatically, the mechanics of its operation here being omitted for the sake of simplified explanation of principles.

An embodiment and application of the invention is in an engine for a motor vehicle, including for example a passenger car, a long-distance truck, a railway locomotive, a bulldozer or grader, etc. Most vehicle engines are designed to operate properly under all conditions reasonably likely to be met using a given specification fuel. The fixed compression/ratio of today's engine is therefore determined by the need to eliminate harmful effects such as pre-detonation or "pinking" under the most difficult operating situation likely to be met - this usually being low engine speed/high load. In fact this situation is only met for a small proportion of engine operation in terms of distance or time, and the engine could digest the specified fuel at considerably higher compression/ratios during the other operating modes. It is known that the power derivable from the combustion of a given amount of fuel increases directly in proportion to increase of compression/ratio, within certain practical limits. It can be seen therefore that the invention, in allowing compression/ratios to vary to be always at optimum for a given running condition allows for the derivation of significantly more

power from a given quantity of fuel, compared with conventional engines, and therefore allows of corresponding fuel economy. As an example, a passenger car engine capable of 6,000 r.p.m. could be constructed to a geometric compression/ratio of 13 to 1, having an optionally and controlledly variable secondary inlet valve lift during compression stroke capable of at most bleeding off a quarter of aspirated charge, the variation being automatically controlled by a combination of engine speed and load. Under the worst situation of low revs/high load (for example starting fully laden up an incline) the bleeding off would be greatest, giving an effective compression/ratio of around 9 to 1 (assuming maximum effective compression/ratio to be 12 to 1). Under intermediate situations such as driving at low speed about town, partial bleeding off would take place, giving an effective ratio of 10.5 to 1, while at highway cruising no bleeding off would occur, giving an effective ratio of 12 to 1, termed 13 to 1 under conventional measurement. As a result overall efficiency and fuel economy, especially under highway cruising, would be much greater than with an equivalent conventionally engined car.

It will have been noted from the foregoing descriptions that a variation in effective compression/ratio will entail a proportionate and corresponding variation in effective capacity. Variation in effective capacity may prove advantageous in a number of situations. For example in the car described above the reduction in effective capacity and consequently fuel consumption is normally high. The reduction in effective capacity need not necessarily mean a proportionate reduction in real (as opposed to theoretical) engine power. For example, if the car's effective compression/ratio/capacity is intended to be reduced at low revs/high load, then under that condition a richer mixture could properly be combusted, probably due to higher b.m.e.p. Thus if a reduction in effective capacity of 10% is made and compensation made by provision of 10% richer mixture, then roughly the same amount of fuel can be burned as in a conventional engine, giving roughly the same power. However, part of the invention specifically consists in the arrangement of the features herein described to provide an

engine intended to allow significant variation of effective capacity. It is known that vehicles are provided with engines of sufficient capacity to provide adequate power under all situations, including emergencies, although this maximum power and therefore capacity is rarely needed or used. It can be seen therefore that the invention, in providing an engine of optionally reducible or variable effective capacity may lead to the achievement of fuel economies. For example, a car may be provided with an engine of maximum effective capacity of 2,000 cc which is optionally and variably reducible to 1,400 cc, the lower range of effective capacity and proportionate fuel consumption being employed during town driving and steady state highway cruising, with the higher capacities only being used at high speed, when overtaking, travelling under load, etc. The variation of capacity may be manually or automatically controlled by any factors, including engine speed, load, ambient atmospheric temperature and pressure, etc. The principles of invention have been described including providing an optimum effective compression/ratio engine and an optimum effective capacity engine. In an ideal embodiment, the invention would be incorporated in an automotive engine to embody and combine in a compromise advantages of both sets of advantages of engine parameter variation.

The principles of the invention may be embodied in an engine in a manner to assist starting. For example, suitable compression/ratios for spark ignition engines tend to increase with speed; therefore during starting and the accompanying low revs, a given engine might require a lower than normal compression/ratio, which could be achieved manually or automatically by variation of valve operation to give maximum reduction in effective ratio during start. In other engines, such as turbo- or super-charged engines, a higher than normal compression/ratio may be desired during start, to compensate for inaction of an engine-dependent blower. In the latter case the invention may be so embodied as to allow charge bleed off during normal running say by secondary valve lift, this secondary lift and bleed off being inoperative during start. In a preferred embodiment, this principle may be incorporated in combination with a

two-stroke engine having secondary lift and bleed off during normal running in order to reduce residual exhaust gas in the combustion volume, bearing in mind that many two-strokes have some kind of assisted aspiration which is engine dependent, and therefore difficult to start. In another preferred embodiment, the two-stroke engine described above may be a turbo- or super-charged marine diesel engine, or other large engines usually operating in the 100 to 1,000 r.p.m. range. Such engines are often installed in today's large commercial cargo ships, such as bulk carriers. super tankers, container ships, etc. It is known that the large single engines these ships may have cannot operate properly below a certain rate of revolutions, and this sometimes causes difficulties in the fine maneuvering of such large vessels. According to another aspect of the invention, an engine is provided with an optional and controlled charge bleed off during slow running, so enabling the power per revolution to be reduced and less mechanical work to be achieved for a given rate of engine revolution. Marine vessels have been cited as a suitable application for an engine where power per revolution is optionally variable, but this principle may be adopted in any kind of specialist or non-specialist engine. A preferred embodiment of the invention is a marine engine having moderate controlled charge bleed off during compression stroke achieved by existing secondary valve lift or by additional valve opening, say to improve charge composition. during "normal" running, this engine having optionally less or no bleed off during starting, and having optionally bleed off to reduce power per revolution, the amount of the latter bleed off being possibly greater than during normal running.

The principles of the invention may also be used to achieve engine braking. One way to achieve this would be to use the features of the invention to optionally and controlledly provide a restricted inlet opening, and/or a large inlet valve secondary opening, so that most of the fluid taken in during the induction stroke would be expelled into the induction system again, leaving the chamber relatively fluid-less when all valves were closed and the piston was at top dead center. This would cause the pistons to work against a relative partial vacuum on the firing stroke (inoperative

with compression ignition engines). Alternatively, the exhaust valve could be arranged under engine braking conditions to have nearly no opening during the exhaust stroke, or to have its major opening during the expansion stroke, either dissipating the force of any firing or in the case of diesel type engines causing exhaust gas to tend to be sucked back in against normal flow. Alternatively, the cam of the invention may be provided with a portion having no or virtually no profile(s), so that under engine braking conditions that portion of the cam would lie opposite the follower, resulting in virtually or completely no valve opening, optionally of both inlet and exhaust valves. If both or all valves were so rendered inoperative during engine braking then the combustion chamber would be sealed, causing a great deal more work to be expended by the piston alternately compressing and expanding the fluid therein, resulting in an increased braking effect. Any or all of these expedients, which involve cam alignments relative to followers which are special to the braking condition, may be actuated by mechanisms or linkages engaged either automatically or by the operator. For example, in large trucks the driver may progressively increase braking effect by moving a lever which actuates the camshaft along its axis or rotation. This could be by introducing an upward movement (say against spring loading) to the throttle pedal, which the driver could actuate by either hooking his foot underneath its surface and levering his foot upwards, or by providing the pedal with an arch (as in racing cycle pedals). In other words a driver would decelerate by lifting his foot off the throttle which would return to idle position, causing a certain amount of engine braking. Further, heavier braking would be caused by upward throttle pedal leverage. Additionally or alternatively, the camshaft movement-actuated heavier braking could be coupled to the first section of the brake pedal depression movement, and actuated by light to medium brake pedal application, as well as heavier application.

As has been noted there is a considerable prior art in variable cam mechanisms, some of the designs involving a cam capable of rotational and axial movement, such cams usually having a conical lobe of progressively increasing cross-

section. Such cams have traditionally involved problems in follower design, for theoretically the follower should have a varying section taken parallel through cam axis, as can be inferred from sections M and N through the cam of Figure 312. The invention consists in the various embodiments described below which are designed to overcome the traditional problems associated with the followers of variable cams.

In order to clarify terminology and assumptions used in this specification there is shown in Figure 312 a typical variable profile cam 30 of the invention, where shaded portion 31 describes the smallest profile and shaded portion 32 the largest profile. Arrow 33 describes the lobe or working portion of the cam, divided into three sections comprising lift portion 34, dwell portion 35, and fall portion 36. In some cam designs the dwell portion is so small as to be virtually non-existent. The non-working portion of the cam is designated by arrow 37. The diagram shows alternatively follower 38 in contact with non-working portion of the cam and follower 39 not in contact with cam, there being a clearance gap 40 between the two. In most rotary-actuated reciprocating mechanisms there is somewhere a clearance and/or adjustment mechanism, so that the follower is only under heavy load during the working portion of the cam movement. In the following description the followers will be assumed to be in light contact with cam during non-working, but the material disclosed is applicable to all types of follower design and location relative to cam during non-working, including being against cam under substantial load. Considering Figure 312, it will be seen that any lateral line or plane of a follower will need to be variously angled in relation to the direction of its reciprocating motion. During the non-working portion of cam movement the plane of follower should be roughly parallel to cam axis, but during the working portion the plane should be variably inclined to cam axis, as is indicated by section N. The non-working portion of cam should in most embodiments be of cylindrical configuration, so that the clearance or loading between cam and follower remain constant with the cam's lateral movement, and consequent variation of profile.

In a preferred embodiment the cam follower comprises a pivotally mounted rotatable roller. In Figure 313 there is shown diagrammatically in perspective a roller 40, mounted on bearings 41 attached to a frame 42 pivotally attached by bearings 43 to reciprocating member 44. The range of roller pivotal motion is indicated by arrows 45. Optionally, there might be provided spring or tension or compression means 46 to bias the roller to say a position perpendicular to member 44 during non-working part of cam movement. For example, there is shown diagrammatically in Figure 314 a twin roller configuration cam follower, having section, taken parallel to cam cross-section, shown diagrammatically in Figure 315, where the roller pivotal centers 47 are connected to saddle member 48 pivotally attached via cup 49 and ball 50 means to reciprocating member 44. Although it is most convenient to relate the movement of two rollers to one reciprocating member, three or more rollers may be linked to one member, or groups of rollers may be linked to several members 44, including in such way as to enable their reciprocating motion to be arranged in sequence, this latter embodiment not being illustrated. The rollers have been shown as cylindrical, but alternatively they may comprise a combination of cylindrical and curved forms to allow for longitudinal curvature of cam profile, such rollers being shown by way of example in Figures 319 to 322.

There is shown in Figure 294 a follower of domed configuration, in many ways the most simple type of follower to be used with a variable profile cam. Because of its apparent single point of contact (in theory all types of follower have an infinitely small point of contact), this type of follower will have greater point loads than other types. However, since it is usually free to rotate, these higher point loads occur over a greater range of locations than other types. Calculations indicate that such types of follower and cam will experience less wear than might be anticipated, this being estimated to be between about two times and five times as great as a conventional type of cam/follower arrangement of the same materials. This smaller than at first apparent rate of wear suggests that it will be commercially viable to embody the invention using

dome-headed followers, providing these are of harder surface than conventional cam assemblies. In another embodiment of the invention it is proposed to modify the dome-head principle to provide what shall be described as a three or more point sledge cam follower, having two or more separate contact areas, such being suitable for cams having profiles of logarithmically or irregularly (rather than linearly) increasing profile, such a cam being illustrated by way of example in Figure 316, where 59 shows in elevation a main lift lobe and 51 in plan a secondary lift lobe. By way of example a multiple point follower 52 having three separate contact areas is shown in diagrammatic plan view in Figure 317 and cross-section in Figure 318, the follower being constructed here in pressed sheet having three domed depressions 53 capable of contact with cam 1 and having between the depressions a pivotal mechanism such as cup 58 to receive ball 57 of reciprocating member 44. In the embodiment shown the sledge will tend to maintain its locations relative to cam direction of rotation 56, since the pulling motion on the total of contact points is twice as great as the pushing motion. However, there might optionally be incorporated elastomeric or compressible members 54 tending to locate the sledge to a desired position relative to cam axis. There has been described a "sledge," but any pivotally mounted member having two or more potential cam contact points may be used as a follower. Contact points of domed cross-section have been shown, but contact points of any cross-section suited to a particular cam configuration may be employed. An alternative contact point is illustrated diagrammatically by way of example in Figures 323 and 324 with reference to the embodiment of Figures 317 and 318. Although followers have been described as having three or more points, two or more of these points may be combined into a linear ridge 55 illustrated by way of example in plan view Figure 325, having cross-section corresponding to Figure 318. It is appreciated that the provision of the followers of Figures 314, 315, and 317 to 322 will involve a more gradual, progressive but earlier reciprocative actuation than with single contact point follower, but any adjustments to compensate for this can easily be made in cam profiles.

There has been described in Figure 295 a non-rotating follower having two or more defined surfaces, each surface corresponding to a specific situation of cam rotation, e.g., to primary lift, to secondary lift, to non-working portion, etc. To ensure optimum performance and wear of both cam and follower it will usually be necessary to prepare the surfaces of non-rotating followers to precise and sophisticated shapes, best described as torsionally distorted flat or curved planes. by way of example, there are described in Figures 326 to 329 in diagrammatic cross-sectional/elevation illustration only, various cam lobes 60 and corresponding follower surface portions 61, where 62 is an intended direction of rotation and 63 direction of reciprocal motion, with shaded portion 64 corresponding to possible follower contact areas for a given lateral position of cam. It will be noted from the diagrams that the reciprocal follower axis need not align with cam rotational axis but may be offset. (It must be noted that the embodiments of Figures 326 to 329 are not shown drawn accurately, but merely to illustrate the principles of follower design involved.) Certain of the followers have been shown having concave surfaces, this having benefits in certain embodiments illustrated by way of example diagrammatically in Figure 330, where 65 is axis of cam rotation in direction 68, 66 axis of reciprocation of concave follower 67. The impact of cam lobe 69 will cause a force to be exerted in the approximate direction 71, while the lift caused at estimated contact point 70 will tend to be in approximate direction 72, loads 71 and 72 arguably according to vector principles combining to load 73 approximately along reciprocation axis. Thus concavity may increase the tendency of contact between cam and follower to be made in front of line of reciprocational axis, such forward contact tending to counteract cam lobe rotational impact loads on follower.

Because higher contact point loads may occur between follower and variable profile cam, the follower may be of bucket 72 type configuration provided with weep holes 71 in the contact surfaces, to permit lubrication in bucket or descending reciprocating member to reach contact surface, as illustrated diagrammatically in Figure 331, where 73 is lubricant. This embodiment may be embodied in any type of follower,

including the sledges having domed surface area. The followers should preferably be capable of some pivotal motion, as may be inferred from the examples illustrated showing ball and cup constructions. If desired, optional restraining means 74 may be incorporated to restrict pivotal motion, say to be operative in one plane only.

There have been disclosed non-rotatable cam followers having separate surface areas, each corresponding to separate cam function. By way of example there is shown in Figure 332 in elevation such a cam follower 81, where 80 is the axis of cam rotation, having a contact area 82 comprising a torsionally distorted curved plane corresponding to primary cam lobe, a contact area 83 comprising a torsionally distorted flat plane corresponding to secondary cam lobe, and an area comprising a non-distorted flat plane 84 corresponding to non-working portion of cam. The borders between the various separated areas may be of any configuration, including of sharp cross-section as in Figure 333, in rounded or progressive cross-section as in Figure 334, or having a cross-section containing a negative depression as in 335, said borders describing the area of any plan configuration. Figures 336 to 338 show by way of example alternative end views of the follower of cross-section shown in Figure 332, where arrow 85 indicates direction of cam lobe approach. The borders between areas are shown clearly defined for simplicity, but may be of any configuration. In certain circumstances, say where clearances are sufficiently accurate, the area corresponding to non-working portion of cam may be eliminated altogether, as shown for example in Figure 339.

The non-rotating follower has been shown of circular cross-section for sake of example but may have a cross-section of any configuration, including rectangular, ovaloid, triangular, of a shape corresponding to a television screen, etc. In an alternative embodiment, a non-rotating follower does not have easily defined areas, but has a generally curved or rounded varying surface. By way of example such a follower

is shown in plan view Figure 340 and in various alternative cross-sections Figures 341 to 344 taken at section lines b, c, d, e, respectively.

It is apparent that the camshafts disclosed herein an in my erroneously abandoned U.S. patent application numbers 565, 654, filed April 7, 1975, may also function as crankshafts. Whether having single or dual function, lateral movement may easily be incorporated in a gas bearing design, as shown schematically in Figure 345, wherein the crank and/or camshaft 86 and its inner main bearing-shell 87 moves laterally inside fixed outer main bearing shell 88. If the diameters of the bearing shells are uniform, then the clearance gap will also be constant, thus maintaining constant gas bearing performance whatever the position of the crank and/or cam shaft. If for some reason it is impractical to move the shaft laterally, the same variable effects can be achieved by interposing a movable yoke as illustrated schematically in plan Figure 346 and cross-section Figure 347. Here the crank and/or cam shaft 89 is fixed, but nevertheless incorporates cams 90 with variable profiles 91. Ball ended and cup ended followers, 92 and 93 respectively, link the cam to appropriate reciprocating mechanisms 94. A yoke 95 is attached to the follower stems 96, preferably by some kind of olive shaped elastomeric washer 97. When the yoke is moved laterally in direction 98, the degree of reciprocating motion in 94 will be varied. Similarly, if the yoke is moved in the other dimension 99, the timing of reciprocating motion relative to cam and/or crank angle will be varied.

As has been disclosed in other sections, cam and/or crank shafts may be supported in variable pressure gas bearings, with gas in the bearing either provided as a gas, or as a liquid conducted under pressure to the clearance space, which then changes state in the lower pressure/higher temperature environment of the clearance space. These fluid pressures may be varied during rotation by what can best be described as moving profile cams, which provide pumping action within the revolving body. In schematic cam/crank section Figure 348, two different arrangements are

shown in a crank disc web 100, having interior passages 101 supplying bearing fluid being interrupted by reservoirs 102 closed by movable plungers 103, the plungers being linked to the free ends of movable pedals 104 pivoted at disc surface 105 and at disc perimeter plane 106. Fixed cam followers 107 are positioned so that when the shaft turns in direction 108. the pedals and therefore plungers are depressed when passing under the followers, causing a pressure wave in the bearing fluid. Such pedal and plunger arrangement can also be adapted to provide engine fuel delivery, where a revolving cam actuates a fixed pedal (not illustrated). If it is desirable that fluid pressure should vary not only with crank rotational angle but also with rotational speed. arrangements similar to that shown schematically in sectional plan Figure 349 and cross-section Figure 350 can be employed. Here a pedal 109 pivoted at 111 is mounted on the disc face of a crank web 110, the pivot being connected to fluid supply 114 and delivery 115 passages. On the external surface of a pedal a weighted shoe 116 is slidably mounted. During rotational movement 117 the shoe will pass under fixed cam follower 118, causing the pedal to be depressed and created a pressure wave in the bearing fluid. The radial motion 119 of the shaped shoe on the surface of the pedal is restrained by spring 120. As rotational speed increases centrifugal force on the shoe will cause the spring to be extended and shoe to move radially outward on the inclined pedal plane, causing the head of the shoe to project further from the disc surface, and increasing plunger motion during each pass under the follower. By such radial movement varied proportionally to centrifugal force, fluid pressure may be varied proportionately to crank revolution speed.

It has been assumed that the reader is aware that in the case of certain internal combustion engines, notably the automotive naturally-aspirated spark-ignition engine, the risk of pre-detonation is a limiting factor in the determination of compression/ratios, and that because detonation is time related the risk of pre-detonation reduces with increase in engine speed. In experiments conducted by, among many others, Roensch and Hughes in 1950 it was found that in terms of a given fuel (98

octane), an engine could run detonation free at 800 r.p.m. at 8 to 1 compression/ratio and at 4,000 r.p.m. at 11.2 to 1 compression/ratio. The benefit in power output over using 8 to 1 compression/ratio at 4,000 r.p.m. amounted to about 11%. Using lower octane fuel (88 octane) the figures were 6 to 1 compression/ratio at 800 r.p.m. and 9.6 to 1 at 4,000 r.p.m.. Here the power benefits were about 20%. Today's engines are generally higher revving than earlier, while octane ratings are being effectively reduced due to emission laws. This suggests that the possible economic benefits of variable compression/ratio engines might be substantial. For example, if it is assumed that such engines, properly developed, were to provide average fuel savings of 15%, then the application of such engines to all automotive units in a country such as the United States would mean that oil imports to that country would be more than halved.

Any or all of the embodiments described in the specification may be used in any combination with each other, and the invention incorporated in any type of engine, in turn incorporated in any type of mechanism or vehicle. For example, the improved engine starting effected by elimination, variation or introduction of charge bleed off process, optionally by provision of secondary valve lift, might also be effected by using the principles of the invention to delay or hasten the opening and/or to vary extent of opening of inlet or exhaust valves. In order to illustrate the principles the cams and followers have generally been shown as solid, but these may be of any materials or construction, including hollow, built-up, of pressed sheet, formed tube, etc., appropriate to any scale of engine, for example from model airplane or lawn mower to giant marine internal combustion engines.

## SIMPLIFICATION

The engines and engine features disclosed in the preceding sections can be further simplified by the considerations and details of embodiments described below. For convenience they are grouped under various headings.

1. **Torroid**

Rather than consider the combustion volume a hollow-cored stub cylinder, it may be perceived as torroidal or doughnut shaped. Figures 351 and 352 show by way of example cross-sections through such combustion chambers, looking toward the cylinder head. If multiple fuel delivery points 2001 are provided in each torroid 2002, then the torroid may be considered a series of abutting chambers 2003 with notional boundaries at 2004. It can be seen that, taking this approach, the total combustion volume can be made as large as desired in a single cylinder application, especially as a feature of the engines of the invention is the drastic reduction of reprocating masses as a design constraint. In other words, the components can be virtually of any size. It is intended that even very large engines, such as for marine and railway applications, can be made in single cylinder configurations. Advantages of the torroidal shape are the relative reduction of surface area and seal lineage per unit volume, a potential reduction of stroke (and therefore piston speed) per unit volume. Table 2 shows how these and other parameters vary with combustion chamber geometry, taking chambers A, B, C, D of Figure 353 as examples. In the diagram, the numbers represent any unit of length, the symbol ø stands for diameter, and engine A depicts a conventional combustion chamber with inlet and exhaust poppet valves. It is assumed that engines B, C, D are the valveless configurations disclosed later in the text. All engines are assumed to have 16:1 compression ratio.

2. **Porting**

It is possible to achieve further simplification by eliminating actuated valves. The interior of the piston/rod assembly can be used for many possible functions, including as a conduit for engine gases, either charge or exhaust or both. Because the piston/rod assembly reciprocates, it is possible to arrange for cross-flow porting. Figure 354 shows schematically such an arrangement, wherein the integral reciprocating piston/rod assembly 2006 moves inside housing 2005, shown here with

torroidal combustion space 2011 at maximum expansion and torroidal combustion space 2012 at maximum compression (the piston as at top/bottom dead center). The rods are hollow containing inlet or exhaust conduits 2008, one of which is shown communicating via exposed ports 2009, the combustion chamber 2011 and exposed ports 2010 with gas handling volume 2013. It is clear that, in this example of a two-stroke engine, a gas flow is induced across the section of the torroidal chamber, and in fact the flow might be in either direction. In the schematic examples of other valveless engines shown in Figures 355, 356, 357 the exhaust and inlet "ends" of the combustion chamber are also interchangeable. Figure 355 shows how inner ports 2009 all communicate with one end 2014 of the reciprocating assembly 2006. Figure 356 shows how the inner ports 2009 for both torroidal combustion chambers 2011, 2012 are served at both ends and are linked by a central passage 2020. Figure 357 shows how the reciprocating piston/rod assembly 2006 can act as a conduit for both inlet and exhaust gas, by for example use of a transfer port at 2015. The inner ports 2009 communicate with a tubular shaped processing volume 2017, which is separated from the other cylindrically shaped engine gas processing volume 2018, which in turn communicates with the transfer ports 2015 by means of openings 2019 and enclosed passages 2016, here shown shaped or tapered for noise reduction purposes.

These kind of valveless embodiments easily permit the introduction of another feature (embodiable with greater complexity in valved engines): multiple varied diameter torroidal combustion chambers which are simultaneously in compression and subsequently expansion and which are shown schematically in Figure 358. Each of the torroidal combustion chambers 2021, 2022, 2023 have the same cross-section but have different diameters. Dimension b represents stroke plus clearance space, while dimension a represents torroid external diameter minus internal diameter. Instead of the three chambers shown, it would be possible to have a single torroid (cross-section b x 3a) of equivalent capacity. However, its clearance space cross-section would be b/cr x 3a, while the cross-section of the clearance space of each of the

torroids shown would be b/cr x a; each would have clearance cross-sectional aspect ratio three times less steep than the single torroid. In other words the multiple chambers shown would be preferable to a single chamber of similar capacity where engines of large capacity and restricted stroke have to be of small overall dimensions. The stepped configurations of the two components also make it easier to design bearing surfaces of the required rigidity. The arrangement shown in Figure 358 permits the two ports to be matched up to each other about midpoint of piston travel, for a relatively brief period relative to porting at bottom dead center (since the piston is travelling at maximum speed). This might be for the purpose of providing extra air to the exhaust, or to cool it. Figure 359 shows an arrangement where there is no such overlap or port to port alignment. Both Figures 358 and 359 are schematic and show only those combustion chambers on one "side" of the piston, that is those chambers that are synchronously all at top or bottom dead center. It is obvious from previous disclosures that additional combustion chamber(s) may be incorporated on the other "side" of the piston.

Such varying diameter coaxial torroidal combustion chambers permit the incorporation of charge processing and other systems within overall engine dimensions, as shown diagrammatically in Figure 359, where 2024 and 2025 are coaxial ancillary systems. Such systems might comprise a supercharger, blower, or impeller, turbo-charger, starter, generator, turbine or other linked engine system. Alternatively the volumes shown at 2024, 2025 might be occupied by systems not directly connected with the engine, such as a liquid or gas pump, rocket motor, ram jet induction or exhaust assembly. Obviously the fixed and moving components can be transposed. For example, in Figures 358 and 359 (which show the synchronous combustion at maximum expansion) component 2006 could be fixed and component 2005 moving. Such an application might be a liquid pumping engine mounted coaxially with or on the pipe carrying the liquid. Generally all the diagrams of this section have been simplified, with fuel delivery, lubrication systems not shown.

3. **Crankshaft**

It will be apparent that the engine configurations disclosed in other sections tend to reduce the effective masses of the reciprocating parts and therefore the stresses that such parts can generate. Engines of a given capacity will tend to have larger and fewer pistons than at present. If only one piston is involved, the variable length piston-to-crank links (of a twin crank layout) can be eliminated if differential crank speeds can be tolerated. (During each revolution one crank has fractionally to slow down or speed up relative to the other to accommodate fixed length links.) In certain applications crank speed variation could be tolerated, for example in an engine powering twin pumps or twin low speed marine screws if the screws have relatively low mass. In other applications constant final drive cycle speeds for each portion of one cycle are required. Various mechanisms can be constructed to convert irregular cycle speed to constant cycle speed. By way of example, Figure 360 shows two crankshafts 2026 connected to a single piston in a cylinder (not shown). They are linked to a final drive 2027 by endless belt, chain or pulley 2028. To compensate for speed variations in the cranks, a movable in direction 2029 carrier and/or variable length tensioner 2030 shortens or lengthens the power transfer distance to the constant cycle speed final drive 2027. The range of movement is indicated by the alternate position of the belt and tensioning rollers 2032 shown dotted, as at 2031. The movement of the carrier may be dampened, as shown at 2033, and need not be reciprocal. It could additionally or alternatively be elliptical, circular, etc. The carrier and/or tensioner may float, positioned by the forces generated in the endless pulley/chain/belt, or it may be controlled by a system of guides and linkages. In schematic partial elevation Figure 361, the spring 2034 loaded tensioner assembly 2030 is mounted about a shaft 2035 (permitting roller 2032 movement in direction 2036) which in turn slidably mounted at one end 2038 on the crankshaft 2026 and slidably mounted at the other end on a fulcrum 2039 fixedly mounted.

Alternatively, if it is acceptable to have the work of compression in one chamber effected by the expansion in the other chamber via the crankshaft (in a twin chamber twin crankshaft layout), then the central reciprocating piston may be split into two halves, with a variable dimension between the halves. Figure 362 illustrates schematically such an embodiment, with piston halves 2040 shown fixedly linked 2041 to crankshafts 2026 shown halfway between bottom and top dead center. Valves, ports, bearings, etc. are not shown. As can be deduced, the distance between the piston halves at mid point of the crank rotation, a, is greater than the distance between the halves at top/bottom dead center, b. The geometry of the engine can be so set up as to make ratio a:b any desired. If the ratio is relatively large, the space between the pistons can be used as an effective compressor or pump (obviously in conjunction with suitable porting, valves by-pass or reservoir volumes and passages), either for the gases of the engine or for non-engine related fluids. In the diagram an energy storage device in the form of a spring 2042 is deployed between the piston halves, to absorb energy at piston travel to top dead center, give it up on travel to pistons' mid position and to again absorb energy on piston travel to bottom dead center. A diaphragm (not shown) may be placed between the piston halves, to divide the volume between the piston half ends. Such multiple volumes can be used to pump or compress separate gases, or by variable relative movement of the diaphragm (and appropriate valving, porting, etc.) to pump fluid from one inter-piston volume to another. Alternatively, the inter-piston volume arrangements described above can be used in a free piston pump or compressor. The basic layout of such an engine would be the same as that of Figure 362, except that the piston rod penetration of the heads, the links 2041 and the cranks 2026 are all eliminated. Such engines will only function properly if the net work being done is regulated to always be less than the actual capability of the engine, for the amount of fuel being supplied. Without such regulation the free piston halves will not return to their designated top dead center position. Of course, with the free piston halves engine, the variation of the inter-piston volume has nothing to do with crank link

geometry; it is instead a function of the pumping or compressing work being done in the volume.

By using an energy absorbing device, such as a spring, the need for a twin crankshaft arrangement can be avoided. To achieve balanced operation of such an engine (having combustion chamber(s) only on one side of the piston), the energy absorbing device in most applications should have a capacity close to half the net work produced by the power stroke (in the case of two stroke engines). Figure 363 shows schematically an energy having one piston/rod assembly 2043, one torroidal combustion chamber 2044, an energy absorbing device 2045 and a volume 2046 in which pumping or compressing work is effected. To simplify all ports, valves, fuel delivery devices have been omitted. Obviously the above principles can be embodied in the engines of Figures 358 and 359.

**4. Cams**

A further simplification can be achieved by eliminating the crankshaft and the fixed or variable length link altogether, instead imparting spin to the piston/rod assembly, which becomes the crankshaft. The spin is achieved by the incorporation of guides, ramps, cams, etc. in such a manner that the reciprocation actuated by combustion is converted into a twisting motion, so that the piston/rod assembly reciprocates and rotates simultaneously. As can be seen from the examples described below, it is generally easier to arrange matters so that several reciprocal cycles are required to complete one piston/rod revolution. In the case of engines operating more effectively at high speed, the lowering of r.p.m. relative to frequency of reciprocation motion (the difference could be an order of magnitude, i.e., tenfold) will enable such engines to be used in a wider range of applications. By varying the reduction ratio, different applications for the same base engine are possible. It is intended that the cam system can be removable and interchangeable in some applications, and that in other applica-

tions there should be two or more cam systems incorporated with one engine, each one of which can be exclusively and selectively engaged, so that such an engine will also function as a variable speed transmission. The cam system, which must at least partly comprise two surfaces which bear on each other at some time (direct contact bearing is not necessary if an air bearing system is used), can also be used to fulfill some other function such as pump or compressor, either to process inlet and/or exhaust gases of the engine or some other fluid such as oil, water, air, etc. The cam system may be incorporated in the combustion chamber(s). For example a torroidal chamber may have part of a surface of sine wave type section. In such case the cam system can comprise a series of separate but communicating combustion chambers arranged to form a sinusoidal torroid.

It will have been noted that the engine of the invention comprises two principal components, the piston/rod assembly and the housing. In the embodiments described earlier one is fixed and the other moves, in the case of engine with the cam system simultaneously in rotating and reciprocating motion. If the housing is mounted in such a way that it may revolve only, then two independently rotating shafts deliver power from a single engine. Such an engine could simultaneously function as a differential and/or be used to power a vehicle, contra-rotating aircraft or marine drives such as propellers, screws, impellers, etc.

Figure 364 illustrates the fundamental principles of one such cam system. A circumferential sine shaped trench 2049 surrounds the midpoint of a piston/rod assembly 2050. In the trench is a guide 2051 fixed to the housing 2052, in such a way that all reciprocating motion is partially converted to rotational motion. Dimension a indicates the broad location of the circumferential band in which the cam system operates. Essentially the cam system is a face cam system, in which the faces are aligned toward directions 2053. When the cam system is referred to as sine shaped it is for convenience, in fact the shape may be of any zig-zag type configuration. There

are certain optimum profiles for each application, shown here within square 2054 which describes one reciprocating cycle. Figure 365 shows the profile of an engine of the type disclosed in Figures 358, 359 which has three cam systems, operating within bands a, b, c. The cam profile for one reciprocal cycle is identical for each band, but a different number of profiles are deployed in series within each circumferential band. The systems described each have a female and a male element (corresponding to trench 2049 and guide 2051 in Figure 364). In the three cam systems of Figure 365 the male elements are wholly or partly retractable, and only those of one band are engaged at any one time. Because loads are alternately transferred from one face to the other, the trench profile need not correspond exactly to the travel path of the piston relative to the housing. The trench might have a clear path 2055, where a small guide will permit piston rotation without reciprocation, and/or a path 2056 which will permit piston reciprocation without rotation. Figure 366 shows schematically a guide of varying size, which may be wholly or partly retractable. It consists of a series of sliding tubes 2057 biased to a retracted position in a housing 2058 and where some hydraulic or other action projects each tube sequentially, those of smallest diameter before those of larger (with retraction effected in reverse sequence). If the smallest form of such a guide is able to describe a clear path in the trench, the arrangement of Figure 365 can be accomplished by having the smallest form all the guides of all three cam systems extended, with selective and/or progressive enlargement of the guides of only one cam system to effect rotation. It is intended that cam systems that have an adjustable portion (e.g., a retractable guide) may also be used as clutches. Without engagement, the engine would only reciprocate; with cam system engagement rotation commences. In the case of guides which are rollers, it may be preferable to have them tapered, with correspondingly inclined cam faces. Figure 367 shows a schematic cross-section through a piston 2059 having axis of rotation at 2060. Two rollers 2061 are fixedly mounted to housing 2062 and rotate about axis 2063 when engaged in channel 2064.

Figure 368 shows schematically a portion of a cam system circumferential band (corresponding to the boxed portion 2054 is Figure 364, but showing a different cam system), wherein the male element or guide 2065 is continuous and has sine wave shaped faces 2066. Axis of rotation is shown at 2067. Trench working faces are shown at 2067a, with system shown solid line in one top/bottom dead center position and dotted line in the other top/bottom dead center position. Kinetic energy will drive the system across bridge a. Such cam systems can be used as pumps or compressors. For example there could be an inlet port at 2068 and outlet port at 2069 and a transfer port 2070 and transfer chamber at 2071 in the case of a compressor. One side of the guide could compress engine charge, the other side could pump exhaust gas, in the case of two-stroke engines, or any other combination of work could be done by the cam system. It is obvious that the cam system could also define at least one torroidal combustion chamber. In the case of Figure 368, two torroidal combustion chambers could be incorporated, with volume 2012 in compression, 2011 in expansion. If the interior of a piston/rod assembly which both rotates and reciprocates is used to deliver charge, and if the ports are not continuous, the design of the interior of the piston/rod and the layout of the ports can be used to spin, slew or swirl the charge into the chamber, if charge movement during ignition is desired. On the other hand, if it is not desired, swirl-less systems such as those illustrated schematically in Figure 369 can be designed (cam system not shown), in which a piston/rod bearing 2072 is attached to the housing 2073.

In the case of a rotating and reciprocating piston/rod assembly half of a generator and/or starter system can be incorporated in the piston or rod, with the other half located in a corresponding position in the housing. Figure 370 shows schematically both a starter/generator having a small circumferential band 2074 on the rod matching with a broader band 2075 in the housing, the bands representing the two portions of the electrical system, as well as a series of elements 2076 mounted on the piston

corresponding to a continuous sinuous circumferential band 2077 mounted in the housing, the band and the series of elements representing a second electrical system.

It is preferable in many applications that the function of the cam system be combined with some kind of pumping or compressing work. Because the cam faces directly or indirectly transfer most of the work that is produced by combustion, it is better (for wear reasons) that no direct contact takes place. The pumping fluid would function as a bearing and heat transfer mechanism.

It is possible, in the case of multiple cam systems, to link the actuation of the guides to completion of all or part of the one reciprocating cycle, actuation simultaneously projecting one and withdrawing one guide.

**5. Drive**

For certain applications, including many pumps and/or compressors, rotary motion is not required. It is both simple and obvious to connect the end of the reciprocating piston/rod assembly to a pumping or compressing device, or to use the variable volume between piston halves as pump or compressor chamber, as disclosed earlier.

However, in many applications it will be preferred for engine final drive to have exclusively rotary motion, requiring a special link between final drive and any reciprocating plus rotary movement of the crankshaft. This can be accomplished by a coupling incorporating either a sliding bearing, such as in a splined propeller shaft, or an assembly incorporating roller, ball needle or taper bearings. By way of example, Figure 371 shows in cross-section and Figure 372 in elevation a schematic of a coupling between a piston/rod assembly 2078 and a final drive shaft 2079 for applications where loads are transferred in one rotational direction only 2080, when roller bearing races

2081 link planes 2082 inside the piston rod and on the shaft 2083. The connection between the two systems could be anywhere, including inside the piston segment of a piston/rod assembly.

Alternatively, the drive may be effected by a bellows type of device, which has rotational stiffness and axial flexibility. Such a bellows device could be of any suitable material, including a spring steel, plastic, ceramic, etc. The bellows device could be one of two broad groups, the closed or sealed type which has an internal variable volume and which might fulfill the additional function of pump or compressor, and the open type which could be considered a series of hinge pairs linked end to end. In many cases energy will be required to deform the bellows. In single piston/twin opposite combustion chamber configurations, it will be preferable if the bellows systems are so deployed that they are in their natural or unloaded position when the piston is in the mid-point of its travel, and that bellows deformation and energy absorption occurs as the piston travel to top/bottom dead center, with stored energy again given up to the piston/rod assembly as it moves toward its mid-point. It is obvious that the energy absorption capability and progression designed into the bellows unit can be used to effect or regulate numerous engine parameters, including variable compression ratio, engine speed, piston breakaway, etc.

If an energy absorption function needs to be incorporated in an alternative drive system, such as concentric splined shafts, then this can be achieved by simple devices, such as a concentric coil spring. In the case of a single combustion chamber engine (or a series of concentric combustion chambers on the same side of the piston), the energy absorbing system could be so deployed that it is in the neutral or unloaded state when the piston is at top dead center, so that the release of stored energy will help accelerate the piston from bottom dead center. In fact the final drive connection could simultaneously function as the main spring or energy storage device affecting the movement of the piston/rod assembly. Figure 373 shows in axial cross-section and

Figure 374 in longitudinal cross-section a discontinuous bellows system. To illustrate different embodiments, two different types of bellows are shown. (Normally only one type would be employed in one system.) At 2084 the bellows is effectively a series of rigid hinges, while at 2085 the same structure defines sealed volumes enclosed by side subsidiary bellows. In similar Figures 375 and 376 a continuous bellows 2086 is shown, defining pumping volume 2087, having non-return valves 2088 permitting gas movement between volume defined by final drive 2089 and volume defined by reciprocating and rotating piston rod 2090.

## 6. Fuel

The reciprocating motion of the piston within the housing (or the housing about the piston) can be used to help regulate the delivery of the fuel. A chamber containing fuel only, or fuel encapsulated by a movable weight can be incorporated in the moving component, in such a way that either centripetal force or the deceleration of the fuel cause a pressure wave to build up in the chamber, which communicates with weep holes opening onto the combustion volume. Figure 377 shows schematically portion of combustion chamber with the piston/rod assembly 2091 just before top dead center (shown dotted). A fuel chamber 2092 in the piston is supplied via supply line 2093 communicating with wick reservoir supply 2094 (disclosed earlier). In the chamber is a weight 2095 restrained by a spring 2096, there being an air pocket 2097 behind the weight, optionally communicating with charge volume by passage 2098. When the piston decelerated toward top dead center the mass of both weight and fuel causes a pressure wave to build up in the fuel, such pressure being sufficient to overcome the surface tension at weep hole 2099 and the pressure in the combustion space. If the piston both reciprocates and rotates, then a separate constant pressure wave can be induced by centripetal forces acting on the mass of the fuel and any optional weight. Mixture can be partly regulated by design of fuel chambers, passages and weep holes. For example, in schematic Figure 378 showing chambers in a

reciprocating and rotating piston, chamber b will supply progressively richer mixture with increase in speed, while in the case of chamber a the mixture will be more constant. There, as speed increases, the pressure increase due to centripetal forces will be compensated for by pressure drop due to deceleration in chamber a. The chambers are shown diagrammatically, and may include the weights, springs, air pockets, disclosed earlier.

In applications where more sophisticated start procedures and equipment can be tolerated, and freedom from maintenance obligations is desired (e.g., aircraft and marine engines, continuously running generators and pumps), then special "componentless" fuel delivery by pressure drop can be used. Generally this will work better if the fuel is superheated and/or is in the system under a given pressure, which can be variable and a function of combustion chamber pressure. At the moment fuel delivery is intended to commence, a local sharp pressure drop in the combustion chamber is induced adjacent to the fuel weep hole, causing the fuel to issue forth. Such a technique may be used to provide all the fuel requirement or only that to initiate combustion in a pre-combustion area. Regulation of quantity of fuel supplied can be by variable restriction of fuel supply passages. Figure 379 shows a piston/rod assembly 2100 having a pocket 2101 which is masked early during the compression stroke. Near top dead center the pocket aligns with a fine passage 2102 communicating with the pre-combustion area 2103 where, because pressure in 2101 is much less than in 2103, a sharp pressure drop is caused, so that fuel issues from chamber 2104, supplied by passage 2105 and optional non-return valve 2106. The pressure in the chamber 2104 may have been linked to that in the combustion chamber by small weep hole and passage 2107. In weep hole type systems there will usually be a volume (it could be a volume containing enough fuel for one combustion cycle under certain operating conditions) just behind the hole, which in turn communicates with the fuel supply. During cold start a small electrical heater deployed in the volume can heat to the desired temperature the fuel delivered for each combustion cycle, with optional variable

heat input to compensate for different speeds during start. Figure 380 shows a precombustion unit 2108 screwed into a housing 2109 by means of keys in slots 2110. Heater 2111 is linked to terminals 2112 in the housing, and fuel delivery volume 2113 is supplied via passage 2114 from fuel delivery galleries 2115.

## FURTHER DEVELOPMENTS

### 1. Central Crankshafts

In the previous section it was shown how the conventional arrangement of a structural element linking reciprocating piston to revolving crankshaft could be eliminated, by imparting spin or turning the reciprocating element. Alternatively, a reciprocating element can be mounted about a crankshaft, as shown schematically in plan section, Figure 381, and elevational section Figure 382, wherein conventional crankshaft 3001 revolving about axis 3002 passes through an elongate slot 3003 in a reciprocating element 3004, in this case communicating with opposed combustion chambers 3005 and 3006 enclosed in a rigidly interconnected housing system 3007, where in operation the inside surfaces 3008 of slot 3003 push on crankpin 3009. For simplicity crankshaft bearings and bearing housings are not shown. It is preferable if the faces of the slot 3008 push on a bearing 3010 (such as a roller bearing) mounted on the crankpin 3009, as shown schematically in Figure 383. Preferably the internal width of slot 3003 should be slightly wider than crankpin or bearing 3010 diameter, providing a clearance gap 3010a. Should energy storage and retrieval systems (such as described elsewhere herein) be desirable, or for other reasons, then elastomeric or compressible elements can be introduced between the reciprocating element and the crankpin, as shown schematically in Figure 384, wherein an elongate sleeve 3011 is mounted within and separated from the elongate slot by elastomeric material 3012, and similarly an outer sleeve 3013 is mounted on and separated from the crankpin bearing

by elastomeric material 3012. In most applications it would be practical to have only one sleeve and elastomeric element.

In selected applications it may be desirable to have a reciprocating element push two coaxial but discrete crankshafts turning in directions opposite to one another, as shown schematically in sectional plan Figure 385 and sectional elevation Figure 386, wherein each crankshaft 3014 terminates in a wheel shaped crankthrow 3015 having a projecting crankpin 3016 and is mounted in a bearing element 3017 in turn mounted in the integral housing system 3007. In this example the crankthrow wheels' circumferences 3018 are restrained by the common idler bearings 3019, and variable two-speed drive is by separately engagable bevel gears 3020, each gear when engaged meshing with concentric toothed rings 3023 integral with crankthrow wheels, so that each crankthrow wheel drives an opposite side of a single bevel at one time. Each bevel is mounted on shaft 3021, the two shafts being slidably mounted on one another. They may be rotationally linked by splines 3022. It can be seen that difference in number of teeth between the outer and inner integral rings on the crankthrow wheel is likely to be smaller than the difference between the number of teeth on the bevel gears, so permitting variation of drive speed with constant crank speed. Here a simple two-speed system is illustrated, but it would be obvious to design more elaborate systems having three or more drive shaft speeds. Also shown in Figure 386 is an optional second system of shafts and bevels 3024 which may be driven by an exhaust gas powered engine or turbine system 3025, putting work into the crankthrow wheels which is in turn taken out by the drive shafts 3021. In the embodiment of Figure 386, the combustion chambers are surrounded by torroidal exhaust gas volumes 3026 which communicate with engine system 3025 via passages, indicated schematically at 3027. Obviously drive can be taken out of or put into the engine via the system linking the twin crankshafts, here bevel gears. The linking system may also serve to transfer work from one crankshaft to the other, so that final drive need only be

connected to one crankshaft. Bearing systems 3017, 3019 are shown by way of example - in individual applications both may not be necessary.

The single reciprocating element/twin combustion chamber units 3028 of Figures 381 and/or 385 can be combined in multiple units either in the same plane as in Figure 387, or in two planes as in Figure 388 or in multiple planes as in Figure 389, wherein 3029 is the first drive shaft assembly and 3030 is a second optional drive shaft assembly. The drive shafts may communicate with transmissions 3031, wheels 3032, propellers 3033 or other systems not shown such as generators, pumps, etc., or with second engines 3034. Where single or multiple units 3028 are mounted about a multiple drive shaft system, each of the shafts may operate at different speeds relative to reciprocation speed by means of the devices disclosed in Figures 385, 386.

**2. Sinusoidal Actuation**

Earlier, Figure 368, a torroidal very roughly sinusoidal combustion chamber was schematically referred to. In fact there were two such chambers separated by what was effectively a flange of very roughly sinusoidal configuration mounted on a reciprocating element. The height of the flange (the dimension parallel to the axis of reciprocation) was shown constant. The shape of the flange (and the heads of the combustion chambers) was not properly sinusoidal; rather the profile approximated a series of straight lines at 90° to each other linked by radius curves. In the case of a reciprocating body turning at constant speed (a desirable objective in the case of engines) a single point on that body will more closely follow a series of circular sine waves, retreating its path every 360°.

Considering one of the inventions in one of its most simplified forms as in Figure 390, we have an upper 3035 and lower 3036 torroidal combustion chamber in an integral housing system 3007 in which a reciprocating element also rotates. The

extreme surface 3037 of each chamber has a similar folded sinusoidal configuration, as sketched in Figure 391, so arranged that the variation of vertical distance between the surfaces is most extreme. The reciprocating element has a flange 3038 which is the working part (it effects compression and transmits expansion forces). The upper and lower surfaces 3039 of the flange are also shaped as in Figure 391, but arranged so that the thickness of the flange is always constant. Because the reciprocating motion is of constant dimension, so the height (distance from peak to valley) of the sine (or similar shaped) wave will be constant, but the pitch (distance from peak to peak) of the wave will vary, from a maximum at the outer radius of the torroidal combustion chamber, to a minimum at the inner radius. Taking a partial cross-section through the two combustion chambers at "A", the path of the reciprocating and rotating flange is sketched in Figure 392, with sine wave height to pitch ration 1:3 and wherein it is assumed that all four sinusoidal surfaces are identical. The path of a fixed point in/on the flange is indicated at several successive positions marked, a, b, etc. The positions of the flange surfaces at corresponding times are indicated 30390a, 3039b, etc. (The intervals correspond to constant units of rotation.) As can be seen, if all four surfaces are identical the engine would not work (e.g., the clearance problem in area B). Therefore, in any one combustion chamber the upper surface of that chamber will have to have a different surface from the lower chamber. Almost any different combination is possible, but in every case it will involve an upper limit on the theoretically possible compression ratio (since the upper and lower surfaces don't match). At the height/ pitch ratio of the sine wave of Figure 393 (1:3), a compression ratio of around 7.5:1 is practicable. If the outermost surface of each chamber retained its sinusoidal form, then a workable form of Section "A" would be as shown schematically in Figure 393. In this case essentially the flange's valleys would stay more or less sinusoidal, but the peaks would have a sharp apex. If the design compression ratio were less than the theoretical maximum, then it would be possible under constant speed operation for the flange apexes to make no contact with surfaces 3037.

Concentric torroidal combustion chambers were mentioned in an earlier section where it was envisaged that they would all be combustion chambers. In fact one or more could be used to compress charge, especially if compression ratios are limited in torroidal combustion chambers. Figure 394 illustrates in schematic half section one such embodiment, whose principals are adaptable to both rotating and non-rotating reciprocating elements - here 3039. A torroidal combustion chamber is shown at 3040 (fully expanded), with compressed charge at "A" moving to displace exhaust at "B." The charge is compressed in chamber 3041 (which may be torroidal or conventional) into which it is conducted via valve 3042, with valve (here poppet type) optionally actuated by some combination of pressure and counterbalance springs 3044. At the end of the compression stroke in chamber 3041, compressed gas enters gas reservoir 3043 via clearance space at "C" and port at "A."

An alternative approach to the "clearance" problem indicated schematically at area B in Figure 392 would be to separate surfaces 3037 from each other, while not increasing flange thickness and therefore separation of surfaces 3039. Such an arrangement is shown schematically in Figure 395. Effectively, this would mean that a point on the flange would no longer describe a sine wave, even though all the surfaces had sine wave shaped cross-sections. The combined rotational and reciprocal motion of the flange 3038 would cause a point on the flange to describe an almost linear, shallowly-curved, "S" shaped path between the apexes of reciprocal motion, with sharp changes of direction at these apexes, compared with a conventional engine, there would be either relatively shorter periods at extremes of pressure or variable rotational speed within one revolution of the flange.

A point on the flange of Figure 393 will describe a sine wave shaped path, this being possible by the creation of clearance space, hereby keeping surfaces 3037 sine wave shaped and making surfaces of the flange irregular. It is obvious that the same effect could have been achieved by doing the opposite - keeping the flange

surfaces sine wave shaped and making surfaces 3037 irregular. Alternatively, a point on the flange could describe a sine wave shaped path, with both the flange surfaces and surfaces 3037 irregular. In this context, irregular means not sine wave shaped.

If the two combustion chambers on each side of a flange are to have a common port system (exhaust or inlet) then the flange will have to be thicker relative to the stroke than is shown in Figures 393, 395. A thickened flange is shown schematically in Figure 396 wherein the combustion chambers 3035, 3036 have the same surface shapes shown in Figure 393. Here a common port system 3045 is located at the outer circumference of the torroidal combustion chambers, with another port system 3046 particular to one chamber located at the inner circumference of the torroid. Obviously chamber 3035 will have an identical port system to 3046 (not shown). If only the flange moves, then port(s) 3045 will be in the fixed components, and port(s) 3046 in the moving flange component(s).

The curves of Figures 392, 393 and 394 are notional and could be said to represent a cross-section taken on a curved plane at constant radius, midway between the outer and inner radius of the torroid, so the ports shown in Figure 396 could be considered projection on this plane, with outer actually larger, inner smaller. In practice, combustion chamber shapes are likely to be a combination of the principles of Figures 393 and 395.

Combined (i.e., both reciprocal and rotational) motion of the "moving" component 3038/3004 relative to the "fixed" component housing 3007 is assumed to be initiated by a starter motor. The shape of surfaces 3037 and 3039 are effective guides to force combined motion, the broadly reciprocal motion caused by combustion being partly translated into rotational motion. Component 3038/3004, having mass, will have both angular momentum and linear momentum. At each cycle the linear moment is substantially absorbed by the work of charge compression, but the angular momentum

is retained by component 3038/3004. Even if the direction of the work of expansion is considered to be parallel to the axis of rotation, angular momentum will cause a point on 3038 to describe a wave shaped path, similar to the shapes of surfaces 3037 and 3039 in the figures. This means that, by adjusting the quantity and distribution of mass in component 3038/3004 and by adjusting the quantity, distribution and/or timing of the combustion, it will be possible under certain operating conditions to so arrange matters that the surfaces need never touch. The natural frequency of motion of component 3038/3004 under those conditions will be such that, during a complete combustion cycle, the surfaces 3037 always (just) clear surfaces 3039. It is desirable for them not to touch for mechanical reasons and because a sinusoidal/torroidal combustion chamber is divided into zones, each zone corresponding to one cycle of the sine or other wave of the surface shapes. The zones of one chamber could be regarded as a series of abutting synchronous combustion chambers, so elimination of surface contact during part of the cycle would permit equalization of gas pressures within the zones and greater mixing of gases.

If such non-contact of surfaces is desired or for other reasons, the combustion process may be tuned by selective placement of the fuel delivery point(s). Figure 395, wherein 3060 shows direction of rotation of component 3038/3004, illustrates conventional-type deployment of fuel delivery points 3047, each here a pre-combustion chamber communicating with a fuel delivery capillary tube, wherein the direction of fuel movement into the main chambers will be roughly parallel to the axis of rotation. Alternative fuel delivery points are also shown in Figure 395 at 4048, where the direction of fuel movement is at a substantial angle, in at least one plane, to the axis of rotation. Theoretically, gas expansion in the main chamber is omni-directional, but in practice the arrangement of 3048 will impart somewhat more rotational movement to component 3038/3004 than the arrangement of 3047, for equal combustion. Figure 396 shows component 3038/3004 with fuel delivery at two locations per combustion zone. Sequential or differential delivery of fuel in the two locations

can be used to regulate the natural movement of 3038/3004. Of course, any number of fuel delivery points per zone may be used. 3049 shows a pre-combustion chamber having a single opening to the main chamber near the mid-point of the sine or other wave, while 3050 shows a similarly located pre-combustion chamber with two openings into the chamber, one larger than the other, and so shaped to give fuel delivery both parallel and angled to the axis of rotation. 3051 shows a similar double-opening pre-combustion chamber with only angled fuel delivery, located at or near the apex of the wave. 3052 and 3053 show single opening chambers at or near the wave apex, with fuel delivery respectively angled to and parallel to the axis of rotation. Any type and combination of fuel delivery locations and directions can be provided in one combustion zone, not necessarily in every zone of one combustion chamber. Any system of fuel delivery can be used, including conventional injectors. The fuel delivery points are shown located in component 3038/3004, but fuel delivery points can additionally or alternatively be in the housing 3007.

In the disclosure, the housing 3007 is described as being fixed. As mentioned earlier in the disclosure, it need not be fixed but can be mounted on bearings inside another housing or enclosure and be free to rotate without reciprocating. Figure 397 illustrates schematically such an arrangement, the rectangles bisected by diagonals representing bearings. A twin torroidal combustion chamber system is represented schematically at 3059. Either because the chambers are sinusoidal and/or because there is a guide system, the combustion process causes component 3004 to both reciprocate and rotate clockwise at a given speed, relative to housing 3056. Component 3004 is linked by splines 3053 to components 3054 and 3055 which are so mounted that they are free to rotate but not reciprocate. They will of course turn in the same direction - clockwise - and speed as component 3004. The housing 3056 is so mounted in an enclosure 3057 so that it is free to rotate but not reciprocate. In practice, if the resistances are balanced, as components 3054 and 3055 turn at, say, 2,000 r.p.m. relative to housing 3056, they will also be turning at around 1,000 r.p.m.

clockwise relative to enclosure, while housing will be turning at around the same speed counter-clockwise relative to the enclosure 3057. Therefore 3054 and 3056 are effectively counter-rotating shafts and surfaces A, B are power take-off points or areas (via gears or friction materials), the assembly being suitable for example in applications such as marine or aircraft having contra-rotating screws or propellers. The speeds of the shafts can be varied relative to enclosure 3057 (but not necessarily relative to each other) by imposition of a resistance indicated schematically by brake pad 3058. In this arrangement component 3055 could be used as a link to another engine system, such as a turbocharger.

A system of concentric co-rotating components may be constructed. Figure 398 indicates one such schematically. Sets of pairs of torroidal combustion chambers of equal cross-section are shown at 3061 through 3064, each set of chambers having progressively smaller radii. Due to combustion chamber design and/or guide systems, the combustion process causes each of components 3065 through 3069 to both reciprocate and rotate relative to its neighbor. Housing 3065 is fixed, the other components all rotate in the same direction relative to housing 3065, and the reciprocating motions are controlled and synchronized through a system of guides so that components 3066 and 3068 reach one apex of reciprocation at the same time as components 3067 and 3069 reach the other apex of reciprocation. The ratio of reciprocations to revolutions need not be the same for each combustion system. Let that of 3061 be 14:1, of 3062 be 11:1, of 3063 be 8:1 and of 3064 be 5:1. If the reciprocations are synchronous, say at 10,000 reciprocations per minute, then component 3066 will revolve at 714·3 r.p.m., component 3067 at 1,623·4 r.p.m., component 3068 at 2873·4 r.p.m., and component 3069 at 4,873·4 r.p.m., all relative to the housing 3065. Component 3069 drives a coaxial turbine system 3070.

A different system of co-rotating components is shown in Figure 399, wherein are four identical systems of torroidal combustion chambers 3079, all sets

having the same reciprocation to revolution ratio. Within a fixed housing 3071 are mounted two components 3073 and 3075 only free to rotate. Concentrically mounted within 3073 and 3075 are another two components 3072 and 3074 able to both rotate and reciprocate, so controlled and synchronized by guides that they simultaneously reach the apexes of reciprocation furthest from each other and simultaneously reach the apexes of reciprocation nearest each other. If component 3072 rotates at 5,000 r.p.m. relative to fixed housing 3071, and all the moving components rotate in the same direction, then components 3073, 3074 and 3075 will turn relative to housing 3071 at speeds of 10,000 r.p.m., 15,000 r.p.m. and 20,000 r.p.m. respectively. Component 3075 drives an element 3078 such as a turbine of a coaxial engine system, with components 3072, 3074 driving (say, via splines) other elements 3080, 3081 of the complex engine at differing speeds. The schematics of Figures 398, 399 are most suited to large high performance and/or high efficiency engines, such as might be used for aircraft propulsion, large marine craft, large scale electric power generation, etc.

It has been indicated above that, by careful component design and regulation of the combustion process, the natural frequency of motion of the reciprocating/revolving component can be such as to enable the wave-like working surfaces of sinusoidal combustion chambers to clear each other. Such design and regulation will be easier to achieve in steady-state engines (for example as used in marine propulsion and generator sets) than in variable-state engines (as used in automobiles and motorcycles). In either case, provision should be made for the natural frequency of motion of the moving component to be varied or disturbed (collision of combustion chamber surfaces should obviously be avoided), even if such variation only occurs infrequently.

In engines with regular (i.e., non-sinusoidal) torroidal combustion chambers, it has been disclosed how reciprocating motion can be translated into combined motion by guide systems. The same kind of guide systems can be used to limit the movement (to just prevent surface contact) of sinusoidal torroidal combustion cham-

bers. For the latter, guide systems can be lighter or fewer than for regular torroidal chambers, where rotational motion is effected by the guides only. The basic mechanical guide system comprises a roller or series of opposed rollers running on endless sinusoidal tracks or in endless sinusoidal groove(s). Figures 400(a) and 400(b) show a typical arrangement, with 400(a) being a schematic plan view and 400(b) the corresponding part elevation, part section, of a six roller system located in a groove (endless) having six waves. The rollers 3084 are shown at one apex of reciprocation, with the opposite apex indicated 3082. The groove housing 3083 is fixedly mounted, while the rollers 3084 are mounted on the reciprocating/rotating component 3004. It will be obvious that the height of the outer perimeter of the groove will have to be greater than the height of the inner perimeter of the groove because the roller has to be cone-shaped, the lines 3085 extending the profile of the cone intersecting the intercept of the axis of rotation of component 3004 and the axes of rotation 3086 of the rollers (one portion of the roller - all of the roller rotating at one speed - has to travel further along the outer perimeter of the groove than another portion of the roller travelling along the inner perimeter; therefore the roller has to have a progressively varying diameter).

Figure 401 shows a roller in a groove, with the roller mounted by roller bearings 3087 on a shaft 3088 which is attached to the housing (not shown) while the groove is located in or mounted on a moving component 3004 (not shown). The groove consists here of three operating parts: an upper track 3089, a lower track 3090 and an optional end track 3091. For better transfer of loads and stresses, the links between these three parts are rounded, and may have optional ventilation holes as at 3092. Obviously, only one side of the roller should be in contact with the groove at any one time, so there has to be some kind of clearance gap 3093. There may be play in the other bearings in the engine system, so the roller has at its end a ball bearing 3094, to prevent the roller from drifting into the groove and so closing the clearance gap. The roller and axle assembly is here retractable from and insertable into the

groove, under some conditions perhaps while there is relative motion between the two, so the roller has a rounded frontal aspect. The working portion of the roller comprises a hard strong engineering material forming the outer casing 3095 (in contact with the groove) mounted on a thin elastomeric intermediate layer 3096 in turn mounted on an inner shell 3097 of engineering material, in turn mounted on the roller bearings. In operation, a load (indicated schematically at 3098) on the roller will cause the shaft 3088 to deflect somewhat, causing the axis of the shaft to become misaligned relative to the track 3089. This misalignment is taken up and absorbed by deformation of the elastomeric material 3096.

It will be obvious that the principles described above can also be embodied by wide separation of the tracks and provision of a second set of rollers. Figure 402 shows component 3004 moving within housing 3007, both defining twin torroidal combustion chambers 3035 and 3036. Two sets of rollers are shown at 3099 and 3100, with the tracks corresponding functionally with those of Figure 401 shown at 3089 and 3090. The relationship of the upper track 3089 to the lower track is assumed to be constant, that is, that the roller during its path along and up and down the groove always maintains the same clearance gap. This condition need not apply. Irrespective of whether the tracks are deployed as in Figure 401 or Figure 402, the relationship of the upper to the lower track may be such that there is a varying clearance gap during one complete combustion cycle. or wave of the groove. Figure 403 shows schematically an elevation taken along the curve of part of a perimeter of a groove, with the line of axis of rotation of the roller shown chain-dotted at 3101. The positions of the upper and lower tracks set out for a constant clearance gap are shown at 3089 and 3090. Possible positions of the tracks consistent with variable clearance gap are shown dotted.

The most useful applications for tracks permitting varying clearance gaps are for engines with variable compression ratios, as disclosed elsewhere herein. In

Figure 403, one track ensures that about the apex of reciprocation a minimum designed compression ratio is reached, while in that region the second track grows more distant from the first, to enable a moving component such as 3004 under certain conditions to travel beyond its designed compression ratio. A symmetrical track separation is shown at 3102, and an asymmetrical track separation at 3103. Variable clearance gap may be desirable for reasons other than variable compression ratio and 3104 shows track separation permitting greater range of component movement around the mid-point of reciprocation. Of course, once there is track separation, the path of the axis of roller rotation can no longer be predicted to always follow line 3101.

The "groove" could be wholly or partly backless (that is, have no end track), permitting gas to pass across the space between upper and lower tracks. Thus the guide system could be located about or within a gas flow associated with combustion. In certain engines it could be in the exhaust flow but generally (because the exhaust gas would tend to pollute the working surfaces and mechanical parts of the guide system), it would be in the charge gas flow. Figure 404 shows schematically a half cross-section of an engine with twin torroidal combustion chambers 3035 and 3036, with the components defining the chambers separated from each other and spaced by the guide components, with both the moving components and the housing components assembled by means of and pre-stressed by tensile members 3105 such as bolts. Here the components enclosing the combustion volumes are of ceramic material, while the guide components are of metal, possibly castings. A single torroidal metal component 3106 containing an endless sinusoidal groove separates identical (but inverted) torroidal components 3110. Charge flow is indicated at 3108, exhaust flow at 3109. Identical torroidal housing components 3111 (inverted relative to each other) are separated by a series of metal components 3112, each having a shaft and roller assembly 3113. Components 3112 have a series of holes 3114 through which charge air passes. Only the curvature of lower track 3090 is indicated; for simplification that of upper track

3089 is omitted. Compressible insulating material is shown at 3110a; ceramic insulation at 3106a.

Other layouts of guide systems relative to combustion chambers are possible. Figure 405 shows schematically a more powerful engine having four identical combustion chambers 3115 and two identical complete guide assemblies 3116, each having upper and lower tracks. The guides have the same number of reciprocations per revolution. and the engine has to be very carefully assembled so that the guides are perfectly synchronous with each other and/or the roller assemblies have to be of the type having elastomeric interlayers. Figure 406 and detail Figure 406a show schematically a twin chamber engine, wherein component 3004 turns clockwise relative to housing 3007. which itself turns counterclockwise relative to enclosure 3120. Three separate complete twin-track torroidal guide systems are located at 3117, 3118, 3119. The sine or other wave in each guide system has the same amplitude but differing pitch, so that each system has a different ratio of reciprocation to rotation. Only one system is engagable at any one time, by means of extensible/retractable roller assemblies. Selection of which guide system is engaged is made by movement of ring 3121 (turning at same speed as housing 3007) by means of actuator(s) 3021a. The ring is connected to a series of slidable shafts or elements 3122 which actuate the extension or retraction of the roller assemblies. Preferably the roller assemblies are spring-loaded to the retracted position. Such retractment/engagement devices are known, but the principle is illustrated schematically for a two-speed system in Figure 406(a), where shaft 3122 has a plate-like section for engagement with head/axle/shaft/roller unit. Sinusoidal tracks may be engagable with non-rotating or solid element, retractable or fixed.

The system of Figure 406 is effectively a machine which combines the functions of internal combustion engine, variable stepped transmission and differential. A clutch function could be located at the interface of the two rotating elements and

any power take-off points. (See also Figure 397). Figure 407 illustrates schematically a machine which combines the functions of internal combustion engine and stepped variable transmission only. Two sets of twin combustion chambers 3115 (four chambers in all) are separated by a power take-off point 3122, in the form of a toothed wheel and shaft, and the transmission system. This comprises three separate guide systems 3123, 3124, 3125, each having upper and lower tracks. The sinusoidal or similar wave system in each guide system has the same amplitude, pitch and curve - they are identical. Because the guide systems are of progressively increasing size, they will have progressively increasing number of cycles, or reciprocations per rotation. As with the system of Figure 406, only one guide system is engagable at any one time. In this arrangement, guide system 3125 represents low gear, 3124 intermediate gear, and 3123 high gear. In another version of this engine/transmission system, the pitches and curves of the guide systems are similar but not identical, each being tuned to combustion and operating characteristics at a particular gear ratio. In variable compression ratio engines, the amplitudes of the guides may also be varied.

### 3. Modular Construction

Multiple concentric combustion chambers of non-uniform size were disclosed earlier herein. They present no theoretical problems of assembly because, as in Figures 358, 359, an integral component 2006 can move and fit within component 2005. It would be useful to have more than two combustion chambers of identical size and configuration, but there would be problems of assembly (especially of the moving component), as can be seen by studying Figure 405. Advantages of being able to combine more than two identical chambers in one engine include the ability to manufacture a range of engines using one set or module of combustion chamber parts.

If one is going to use one combustion chamber module to make engines of varying power and swept volume, then the gas passage(s) within the module (if any)

should be so sized as to accommodate the gas flows of the largest engines likely to use that module. Figures 408 to 411 illustrate schematically various possible gas flow layouts, wherein 3126 indicates a multiplicity of equal sized torroidal combustion chambers, 3004 the moving component, 3007 the "fixed" housing (which, in all these embodiments, could also rotate), 3057 an enclosure or casing. A represents charge air volume, B high temperature and pressure exhaust, C lower temperature/pressure exhaust. Porting is not shown, but can be as described elsewhere in this disclosure. Solid arrows describe gas flows through ports, dotted arrows show gas flow to and/or from transfer ports or flows via passage or plenums as described elsewhere herein. Insulation is indicated (schematically, like all other components) at 3127. In Figure 408, insulation separates charge flow from hot components, charge flows into the combustion chamber, exhaust flow from it into a central exhaust gas reservoir. Obviously the flows could be reversed, volumes A and B transposed, insulation moved to the interface of component 3004 and the central (now charge) gas reservoir or plenum. Figure 409 shows a system having transfer ports, indicated schematically at 3128. Here again, the flows could be reversed, volumes transposed, insulation repositioned. Figure 410 shows a layout where exhaust gas flows adjacent to the structural component of 3004 and 3007 are used to reduce heat flows (i.e., thermal gradients) across these components, with the center of the engine occupied by a mechanical system 3130. If 3130 were a fuel delivery system, this could serve to maintain liquid fuel under pressure at temperatures greater than boiling. A compressor/turbine system is indicated at 3129/3134.

A preferred embodiment is indicated in Figure 411. Here, ambient air enters a compressor 3129 at 3136, high pressure charge is delivered via plenum or anulus 3131 to tubular volume A, in which heat exchangers 3132 are located for purposes of after-cooling. Optionally water under pressure circulates in the heat exchangers, to be used for compounding (as described later) and /or to provide bearing pressure as disclosed earlier. Hot exhaust from tubular volume B goes via plenum or

anulus 3133 to turbine 3134, which is mechanically linked to the compressor 3129. On leaving the turbine, the lower temperature exhaust flows through tubular volume C to exit at 3135. If the number of equal concentric torroidal chambers at 3126 is relatively large, then the engine will or might have a torpedo-like or tubular shape. This, together with the uni-directional gas flows indicated at 3136 and 3135, will make such engines suited for particular applications, as in aircraft or certain marine craft. Obviously, additional compounding can extract further work from the lower temperature exhaust gas at or after 3135. In certain embodiments, the separate insulation 3127 need not be employed, especially if the gas flows are large per unit volume and/or the structural components used in 3004 and 3007 have moderate to good insulating properties.

A schematic profile of a half cross-section of the torroidal form of a typical combustion chamber is shown in figure 412. It is drawn with the ratio of height H to outer radius R2 minus inner radius R1 equal to 6:5. Here the maximum inlet port is shown at 3137, the maximum exhaust port opening at 3238, with dimensions I and E being 0·183 x H and 0·267 x H respectively. If the motion of 3004 relative to 3007 is represented by the sine curve, then the port/valve openings, measured in crank angle from top dead center, are: exhaust opens 114·7, inlet opens 126·9°, inlet closes 233·1°, exhaust closes 245·3. If the ratio of (R2 - R1) to R1 is 1:2·5, the ratio of maximum inlet port area to maximum exhaust port area is 1:1·04. Dimension S represents the stroke. The working surfaces A and B are angled relative to cylinder walls C and D as shown, and the intercepts of the surfaces are rounded as shown. so that the gas flows across the combustion volume are as smooth as possible and so that stresses are reduced and more evenly distributed in monolithic components 3004 and 3007. The object of the smooth gas flows is to optimize two-stroke scavenging and minimize residual exhaust gas left in the charge after the ports close. The chamber is shown at maximum volume; component 3004 will move in direction 3139 to effect compression.

Figure 416 shows by way of example an engine assembly whose combustion chambers are of modular construction, wherein details A and B are half vertical sections along the different radii indicated in details C, D and E, which are cross-sections through the planes indicated in the vertical sections. Component 3004 at least reciprocates relative to component 3007 and is shown at bottom dead center. Details C, D and E are shown with components 3004 and 3007 in different positions relative to each other, when the appropriate detail lines shown on the vertical sections are in alignment with each other. Identical ceramic reciprocating components are shown at 3155, with identical ceramic "housing" components shown at 3156. Charge circulates through volume 3157 and enters combustion chambers 3126 via inlet ports 3158, exits via exhaust ports 3159. Exhaust gas circulates through tubular volume 3160 and is spaced from outer enclosure 3057 by insulation 3127. Exhaust gas circulates to some degree within spaces 3161, 3162. Since these communicate with the main exhaust gas circulation volume 3160, they serve to reduce thermal gradients in selected portions of the combustion chamber components. A gas bearing supplied by super-heated liquid is shown, schematically, at 3163... The respective components are assembled and fastened (preferably pre-loaded in compression) by means of tensile fasteners 3164 and 3165. Fasteners 3164 are located within the relatively cool charge flow volume and so are of conventional design, while fasteners 3165 are adjacent hot components 3156 (separating hot combustion chamber and hot exhaust volumes) and so are of tubular design. the interior of the tube communicating with cooler volumes (say charge) this circulation of cooler gases through the interior of the fasteners serving to maintain their temperatures below the temperatures of components 3156. Loads are distributed along the rims or extremities of components 3155 and 3156 by means of load distributor elements 3166. 3167, 3168, 3169 which, in preferred embodiments have additional other function(s) including possibly guide system, electric motor/generator, bearing and/or sealing components. They may also function as fuel delivery system or tribology system components. The matter of tribology and bearings as well as sealing will be described later in the disclosure. Figure 417 shows a cross-section detail of an optional alterna-

tive to fastener 3165, wherein hollow tensile member 3170 does not fit tightly within component(s) 3156 but is separated by an insulating and/or elastomeric interlayer 3171, which cold be of any suitable material, including ceramic wool. The engine illustrated in Figure 416 has four identical combustion chambers. It is obvious that other engines using components 3155 and 3156 can be constructed, including ones having two combustion chambers and, if volumes 3157 and 3160 are sufficiently large, engines with six or even more combustion chambers. Alternatively, components 3157 and 3160 can be used in other engines with four combustion chambers, for example, wherein heat exchanges are located within volume 3160 and the enclosure 3057 is therefore of large diameter. When constructing different engines using standard components 3155 and 3156, it is probable that other components such as the fasteners, enclosures, etc. will differ and be particular to each engine design. The combustion chambers illustrated in Figure 416 and elsewhere generally show an angle between wall and head/crown (angle Ø in Figure 422) of around 110° to 120°. In fact, the chambers could be designed with Ø any suitable angle, including 90°.

Figures 418, 419 and 420 show further examples of engines having combustion chambers of modular construction. The method of illustration is similar to that of Figure 416 (Figures 418, 419 and 420 each show a different engine), and both the size/configuration of the combustion chambers and the basic configuration of torroidal components 3155 and 3156 are similar in all four engines. Variations occur mainly in the gas flows and the methods by which loads to and from components 3155 and 3156 are transmitted. Because Figures 418 and 419 illustrate how two substantially different engines can be assembled using the same combustion chamber components, the details A, B, C, D and E of each figure are presented side by side, for purposes of comparison. Combustion chamber components 3155 and 3156, as well as the cross-section of fasteners 3164 (but not necessarily their length) are identical in both engines. Insulation 3127 is deployed as indicated in both engines, as are load distributor elements 3166, 3167, 3168, 3169.

In the engine of Figure 418, charge air circulates in tubular volume 3172, enters the combustion chambers via inlet port 3173, exits via exhaust port 3174 into high temperature/pressure exhaust gas circulation volume 3175. The approximate shape of this volume resembles a series of stubby cylinders, one linked to the other by multiple tubes, the tubes being deployed along a circumference approximately equal to the mean cylinder circumference, neither cylindrical or tubular shapes being regular. The exhaust gas passes to a turbocharger (not shown; the layout of Figure 411 would be suitable), and from there low temperature/pressure exhaust gas passes down the central volume 3176. Components 3155 are separated from each other and the load distributor elements by spacer rings 3177 and spacer plates 3178 having holes to accommodate volume 3175. Components 3156 are separated from each other and the load distributor elements by spacer rings 3179, each having a series of internal projections (see illustrations), and by inlet port rings 3180, each ring having a series of holes permitting the passage of charge air (see illustrations). Here the ring comprises an integral element having an upper rail and a lower rail separated by a series of posts (which accommodate the fasteners 3164), the transitions between them being rounded and smoothed. The tubular charge volume 3172 is enclosed by a casing 3181, here having within it passages 3182 containing circulating liquid, for the purpose of cooling the casing and therefore indirectly the charge.

The engine of Figure 419 has the same combustion chamber components 3155 and 3156 as that of Figure 418, and is therefore presumed to have the same stroke and inlet and exhaust port openings, ports shown at 3173 and 3174, respectively. However, the gas flow is different, charge flowing in central volume 3183 to the inlet port via passages 3184 and transfer part 3185. The difference from the engine of Figure 418 has been achieved only by substituting spacer plate(s) 3178 with a series of eight smaller ring-shaped spacer plates 3186, each also able to accommodate volume 3175, and by substituting the inlet port ring(s) 3180 with transfer port rings 3187. Note that spacer elements 3177 and 3179 remain unchanged. Since the gas flows are

different, outer casing 3181 can be eliminated. In both engines there are located within or adjacent to components 3155 and 3156 special volumes 3188 which communicate with volume 3175 and will therefore also contain exhaust gas. As previously, the objective of volumes 3188 is to reduce combustion chamber heat loss through components 3155 and 3156.

The engine of Figure 420 illustrates alternate ways of assembling/fastening/mounting modular combustion chamber components. Components 3189 and 3190 are similar tho those illustrated previously, as are volumes 3188 housing or permitting the passage of exhaust gas. Here charge travels within tubular volume 3172 via inlet port 3173 to the combustion chamber; exhaust exits via exhaust port 3174 to central tubular exhaust gas volume 3191. Instead of using conventional tensile fasteners (such as 3164 in Figures 418, 419), this engine is assembled by means of pierced tubes. Inner tube 3192 is continuously threaded on its outer surface. Load distribution rings(s) 3193 are threaded onto the inner tube 3192, and once in final position secured by means of locator pins or keys 3194. The rings support components 3189, which are further restrained by sleeves 3195 of rectangular form with rounded corners inserted into pre-formed holes in tube 3192, and restrained by means of pins 3196. Exhaust gas passes from port 3174 through this sleeve 3195 to volume 3191. In a similar manner, components 3190 are supported by means of load distribution ring(s) 3197 threaded within outer tube 3198, and when in final position secured by means of locator pins or keys 3194. Components 3190 are further restrained by circular sleeves 3199 threaded into pre-formed holes in outer tube 3198 and restrained by means of pins 3196. Inlet charge passes from volume 3172 through this sleeve 3199 to inlet port 3173. Insulation 3127 within and against outer tube 3198 prevents heat loss from exhaust gas in volumes 3188. An outer casing 3181 defines volume 3172. In an alternative embodiment, illustrated only in details B and E, the casing has a multiplicity of projections 3200 located in the charge air flow, and is made of material having good thermal conductivity, for the purpose of transferring heat from the charge to beyond the casing 3181

(a form of after-cooling). This device is particularly useful in situations where the fluid surrounding the casings is at low temperature, say under water in marine applications or at high altitude in aircraft application. The projections 3200 are shown schematically only; they can be of any configuration and integral with the casing or attached to it in any way. Exhaust gas reaches volumes 3188 associated with components 3189 from volume 3191 via holes 3201 in inner tube 3192, which is of varying thickness in cross-section, stiffening ribs 3202 running vertically or longitudinally on the inside of the tube between the exhaust sleeves 3195. Within each rib are two capillary fuel tubes 3203 (one to supply all the chambers moving 3004 in one direction, the other for the chambers moving 3004 in the other direction), which communicate with the combustion chamber via load distribution ring(s) 3193. Here, two tubes 3203 are shown in each longitudinal rib, however any twin system of tubes and/or galleries may be used, supplying the chambers via ring(s) 3193 and/or directly. The fuel supply need not be within the tube, but could be in fuel lines within volume 3191 to pierce 3192 via connectors, couplings, etc. Fuel delivery is here shown associated with the inner tube; it could be equally associated with the outer tube 3198. A similar system of tubes/passages/fuel lines could be used to provide fuel used for tribological purposes to any desired location within the engine. In Figures 418, 419, the fasteners were attached to load distributor elements 3166 to 3169. Here, the outermost rings 3197 could be identical to an inner ring 3197, or they could be integral with a component 3204 having another function, such as bearing, gas-seal, guide system element, electric motor/generator component, as indicated in the diagram. To prevent differential rotation between components 3189/3190 and their respective support rings 3193/3197, the support surfaces of the rings may have projections and/or undulations matching indentations or undulations on the corresponding support surfaces of the combustion chamber components. In schematic illustration, Figure 421 shows elevationally part of a ring having support surface undulations, while Figure 422 similarly shows part of a ring having projections or nipples which also have fuel delivery tubes.

The engines of Figures 418, 419 and 420 all show tensile fasteners of circular cross-section arranged parallel to the axis of reciprocation. The fasteners could have any appropriate cross-section, including that of straps or thin strips of sheet, arranged at any angle to the axis of reciprocation. Figure 423 shows very schematically a system of strap-like fasteners arranged at angle to and constant radius from the axis of reciprocation. Of course, in most applications there would have to be a second and corresponding system of fasteners angled either in the opposite direction or parallel to the axis of rotation. In the case of a housing or casing of cylindrical shape having internal passages, say for cooling fluid, these passages could also run mainly diagonally, as shown very schematically in Figure 424 and implied by the details and sections of Figure 418. Similarly, where tubes are used structurally (as in 3192, 3198 in Figure 420), any apertures in such tubes could be of any shape and/or direction, including diagonally. In the case of straps running on a curve (as illustrated in Figure 423), or in the case of either thin-wall tubes, or tubes with cut-outs whose primary dimension is not parallel to the axis of reciprocation, then such straps and/or tubes will probably have to be restrained. Usually the most practical form of restraint will be the torroidal compulsion chamber components likely to be within them. In acting as such restraint, the combustion chamber components would be loaded in compression radially inwardly toward and more or less perpendicularly to the axis of reciprocation. Therefore, the provision of the straps or the thin-wall or other tubes mentioned immediately above could be used as a design tool to distribute or create desired loadings in the combustion chamber components.

Fuel delivery passages have been generally shown equal to each other and travelling in a series of straight lines. They need not be equal nor be linear. In the case of several fuel delivery points being supplied from a common fuel delivery reservoir or gallery, it may be desirable to have equal delivery path lengths although the delivery points are unequally spaced from the reservoir or gallery. In such case the

arrangement of Figure 425 can be considered, wherein 3205 are fuel delivery points, 3206 equal length passages, 3207 a gallery all arranged within a tube 3208.

The modular combustion chamber layouts of Figures 410 through 420 have been designed to be used for engines wherein component assembly 3004 only reciprocates, or it both reciprocates and rotates. According to which, the function of the components such as 3166 to 3169, 3204 attached to the structural elements (such as fasteners, tubes) will vary, either being linked to guide systems or to some kind of crankshaft. The combustion chambers are assumed to be of regular torroidal configuration, but the concepts and sections could be applied equally to sinusoidal torroidal combustion chambers, should both compound motion of 3004 and a relatively lightly stressed guide system be desired.

All the components shown in Figures 416 through 420 can be constructed of any suitable material. Generally it will be preferred that the combustion chamber components 3155, 3156, 3189, 3190 be of ceramic material, while the fastening or structural components 3164, 3165, 3192, 3198 be of metallic material. Components 3180 (inlet port ring) and 3187 (transfer port ring) could be suitably constructed of ceramic or metal (as well as other material). It will perhaps be preferable for other spacer components to be of ceramic material.

For the sake of simplification the components have been shown abutting each other. In fact, any kind of suitable interlayers or materials could be used (the interlayers are not illustrated generally in Figures 416 to 420), including gaskets, ceramic wool, etc. In a preferred embodiment, components are coated with a powder, say by electrostatic deposition, prior to assembly, which remains as a very thin spacer between components after final assembly. The composition of the powder may be such as to cause it to slowly bond to one or another of the component with increased engine use and subjectment to heat and cooling.

The different concepts in this disclosure can be combined in any way. For example, any single combustion chamber can be deployed each side of a guide system, a conventional crankshaft or a slot-drive crankshaft (as shown in Figures 381 to 386, for example). Any combination of combustion chambers can be arranged each side of the above mentioned drive or guide devices, the numbers of the chambers and their configuration not necessarily being the same on each side. In a further example, the combustion chamber grouping of Figure 358 can be arranged on one side or either side of a different drive system, the power take-off (Figure 407), with separate retractable guide systems associated with each of the differently sized chambers, either the largest or smallest chamber closest to the drive, to provide engines having three or six torroidal combustion chambers of three different sizes. In a further example, the combustion chamber and pumping chamber combination of Figure 394 can be arranged on one or both sides of a slot-drive crankshaft, such as in Figures 381 through 386. Where reference is made to a slot-drive crankshaft, any type of slot-drive crankshaft system may be used, including the single crank device known as a scotch yoke. Generally, it will be sensible to group combustion chambers in coaxial pairs, with each of a pair on opposite sides of a central flange forming part of a reciprocating system, and/or each side of a more or less centrally located guide system(s), conventional or slot-drive crankshaft(s). However, multiple chambers need not be either equal or coaxial, and could be deployed in any fashion about a crankshaft or other drive or guide system, as shown for example in Figure 389. Where appropriate, "sinusoidal" torroidal chambers may be used (such as are shown in Figures 390 through 396, for example), instead of the "regular" torroidal chambers generally illustrated. The "regular" torroidal chambers may be defined as surrounding or containing within them a component which just reciprocates or which both reciprocates and is caused to rotate by a guide system. "Sinusoidal" torroidal chambers may be defined as having surfaces which are not on a plane but have substantial three dimensional form of regular configuration. By regular, it is meant that an entire surface has a form consisting of a sub-form which repeats (but the sub-form may also comprise the whole form in special

cases), this sub-form (or whole form) having a wave-like configuration, the wave being defined by the sine curve or any other mathematical formula. Here, wave form is meant to include a series of apexes linked by straight lines or planes. The slot drive crankshafts can be used singly in any location (for example, they could replace the guide systems in Figures 405 to 407) or they can be used in multiples, as shown schematically in Figures 189 to 192 where they would replace the regular crankshaft. Torroidal combustion chambers or pumping volumes can be used in combination with non-torroidal combustion or pumping chambers, as shown schematically in Figure 413, wherein a piston-like component 3140 drives a crank 3141 via a slot 3142. The regularly shaped working volume 3143 and the torroidally shaped working volume 3144 can each be either a combustion chamber or a pumping chamber. If both are pumping chambers, work can be put into the system via the crank 3141. There need be no crank or guide or any drive system. Figure 414 shows an arrangement wherein a torroidal combustion chamber 3146 drives a piston 3145 which works a pumping volume 3147. In operation, combustion chamber expansion causes pumped fluid to exit volume 3147 in direction 3148 via non-return valve 3149, and combustion chamber compression is effected by pressure from fluid entering volume 3147 from direction 3150 via non-return valve 3151. (Such a machine could be used to give pressure boost in pipe flows.) Generally in this disclosure, like numbered parts have similar characteristics and/or functions.

Combustion chambers may be separated (singly or in groups) by mechanical systems other than those described above - conventional crankshafts, slot-drive crankshafts and guide systems. For example, combustion chambers can be separated by an electric motor or electric generator. If component 3004 has compound motion, the windings of the electrical assembly need not have the conventional band-form but could have a sinusoidal torroidal form, the shape of the sine-like wave of the electrical winding corresponding to the motion of a point on 3004. As an alternative to placing electrical motors and/or generators between combustion chambers, such electrical

systems could be placed on the side of multiple concentric torroidal combustion chambers. As electrical machines can be deployed in this way, so can other machines or mechanical devices, including the following: pumps, compressors (both of either torroidal or other configuration), counting devices speedometer drives, power take-off points, transmissions, clutches, fuel delivery machines or pumps, lubrication machines or pumps, machines or pumps associated with inter-cooling, engines employed to extract additional work out of the exhaust gas (that is, for compounding), etc. In Figure 397 there is shown schematically by way of example the windings 3058a of an electric starter motor/generator, one set of windings located on component 3056, another set on component 3055, electric performance being related to the rotational motion of 3055 relative to 3056. Figure 415 shows schematically, by way of example, two pairs of combustion chambers 3126 separated both by a guide system 3152 and an electric generator/starter motor. 3153 shows a sinusoidal torroidal winding integral with "housing" component 3007, while 3154 shows the other regular torroidal winding integral with component 3004. The location of the sinusoidal winding relative to the regularly shaped winding could be reversed - the sinusoidal winding being on component 3004 (not shown). In some embodiments the same components could at least partly function simultaneously as guide system and electrical motor and/or generator.

## 4. New Marine Craft

As has been suggested previously in this text, and shown schematically in Figures 260 to 263, new high power-to-weight ratio engines would permit construction of marine craft which in normal operation would have their hulls out of the water attached via post(s) to one or more engines plus control and stabilizing devices in the water for such craft to be practical, certain features are desirable. It must be efficiently and safely operable when for certain reasons (e.g., weather) the hull cannot be lifted above the water. If operable at moderate or greater speeds with the hull above the water, then those parts of the craft which remain below the water surface should have

a streamlined aspect with swept-back projections, to reduce risk of snagging lines and better deflect encountered objects, as well as to present the least possible drag. The distance between the bottom of the main portion of the hull (when operable in the normal mode), and the surface of the water should be sufficient to permit the hull to fully clear waves/chop of a designated amplitude while still maintaining the in-the-water elements at a minimum depth below the variable surface at all times. This distance, together with the height/depth of the in-the-water elements, is likely to be such as to give the craft a relatively deep draft when operated with the hull in the wake or at anchor. If such a craft is to have a reasonable shallow-water capability, then the in-the-water elements should be retractable/extendible relative to the hull.

Figure 426 shows in schematic elevation a craft having the basic elements to support the optional movement of the hull 4001 above the water line 4002 in direction 4003, comprising a retractable post 4004 attached to a keel or keel-like element 4005, to which hydrofoils or hydrofoil elements 4006 and at least one adjustable vertical hydrofoil or rudder 4007 are attached. Figure 427 shows in schematic frontal elevation the craft of Figure 426, having a mono-post. Figures 428, 429 show frontal elevations of different craft, which have a longitudinal elevation similar to the craft of Figure 415. In Figure 428 that craft has twin attached extensible/retractable posts, and in Figure 429, that craft has twin non-attached extensible/retractable posts. In the examples which follow generally extensible/retractable posts will be shown, but in the case of craft not requiring shallow water capability, the posts may be fixed (i.e., non-retractable/extensible). The principles of the invention apply to both types of post. A single base vessel design may have a particular post-system, but have varying keel element(s) for particular embodiments of the base design. The hydrofoils or hydrofoil elements may be fixedly mounted, so that they will support at a given elevation at a particular speed/weather combination, or they may be pivotally or otherwise variably mounted to present variable frontal aspect and therefore give variable lift at a particular speed/weather combination. Generally, variable deflection/lift hydrofoil elements

will be shown by way of example in this text, but the principles of the invention also apply to fixedly mounted hydrofoil elements.

In mechanically powered craft at least one drive means such as a propeller or water jet is preferably incorporated in the keel element, although it could alternatively be mounted in/on the post or hydrofoil element. In the case of craft which do not have posts mounted behind each other, each keel element/post combination should have at least two sets of hydrofoil elements mounted one aft of the other and preferably also a rudder device to facilitate lower speed directional control. (At high speed, some measure of directional control can be achieved by varying the angle of attack of the hydrofoil(s) on one side of the boat or craft from that of the hydrofoils on the other, effectively banking the craft. Alternate methods of directional control in the cases of the craft of Figures 428, 429 is to vary the power provided to the drives in each keel element.) Figure 430 illustrates schematically how post and keel elements can be mounted behind each other, either to give a total of two assemblies, or if paired as in Figures 428, 429, a total of four assemblies. The engine(s) which power the drive means may be mounted in the keel element(s), the post(s) or the hull. The principles of the invention apply to the mono-hulls illustrated by way of example as well as to multi-hulls such as catamarans or trimarans. The principles of the invention apply also to craft not mechanically driven, that is to wind driven or solar powered craft. By way of example, mechanically driven craft are shown, some of which may also have sails. The craft which have sails will more likely operate under mechanical power with the hull out of the water and under sail with the hull in the water. In the case of some craft with sails (with or without engines), under certain specific conditions of water, wind force, real wind direction, apparent wind direction and through-the-water speed, the devices of the invention may be used to propel the craft by wind with the hull wholly or partly out of the water, whether or not an engine has been used to attain that condition. In Figure 426, 4002a shows waterline with hull in water.

The twin keel elements of Figures 428, 429 are shown having parallel vertical center-line planes (whether or not these planes align with the axes of their respective posts), but they may also have angled or splayed planes which are on an angle other than 90° to the mean water surface - that is non-perpendicular. In the case of keel elements having drive means 4008 mounted forward of a rear-mounted rudder element 4007, the drive means can be mounted between, above or below those keel element portion(s) supporting the rudder, as shown respectively in Figures 431, 432, 433. In the case of the arrangement of Figure 430 with keel/post assemblies mounted fore/aft of each other, the keel elements may be truncated longitudinally, have only one set of hydrofoil assemblies each and may be pivotally mounted above axis line 4007a on the post to also comprise a rudder or steering means. By way of example, a front keel element is fixedly mounted and has no drive means, while a rear keel element is both pivotally mounted and has a drive means. Alternatively, the drive means could be incorporated in a fixedly-mounted rear keel element while the drive-less front keel element is pivotally mounted on the post. Where power requirements are great both front and rear keel elements could have drive means. Figure 434 shows a smaller keel assembly 4005 mounted on a smaller post 4004, suitable for the craft of say Figure 430 (rudder device omitted for simplification). The keel element may house a snap-in engine module 4009, as shown in alternate positions in Figures 435, 436, 437. (Line 4007a shows optional rudder function.)

The hydrofoil elements including the rudder element may be wholly or partly extensible/retractable. By way of example, a single hydrofoil element is shown mounted on a keel element 4005 in Figures 438 through 443, wherein the general arrangement is shown in plan view Figure 438, frontal elevation Figure 439, sectional elevation A-A in Figure 440, sectional elevation B-B in Figure 442, detail C enlargement in Figure 442, detail D enlargement in Figure 443. In this embodiment the main hydrofoil body 4010 and its extension 4011 are of approximately symmetrical cross-section and are adjustably pivotable about 4012. In other embodiments non-symmetri-

cal or wing-section hydrofoils may be used. In fact the angle of attack (or degree of pivot) is unlikely to be more than around 12 degrees each side of the neutral or "straight ahead" position, but is shown at exaggerated angle at 4012a in Figure 440 for illustrative purpose. The main hydrofoil body 4010 is integrally mounted on a bearing disc 4013, its inner face attached to actuation points 4014 terminating actuation levers or tensile elements (not shown). The inner face of disc 4013 also has a connection point 4015 for the system activating the motion of hydrofoil extension 4011. Any suitable system may be employed, including the one illustrated schematically here, wherein hydraulic fluid coupled via connector 4015 activates movement of a piston 4016 (housed within hydrofoil 4010) tightly slidably mounted within a cylindrical recess 4017 within hydrofoil extension 4011. A hollow passage 4018 conducts hydraulic fluid to and from variable volume fluid reservoir 4019 in cylindrical recess 4017. Fluid sealing devices are indicated schematically at 4020. A coil spring 4021 is mounted about the piston to facilitate the retraction of 4011 within 4010 in the event of loss of hydraulic power or under other circumstances. The anchored end of 4011 has a scalloped profile 4022 to distribute shear stress induced in 4010 by upthrust or downthrust on 4011. The bearing disc 4013 has a stepped cross-sectional profile at its perimeter, seating in a corresponding profile in the keel section, and is retained by a removable rung 4023. The junction between the bearing disc, the keel section and the ring includes bearing surfaces, optional bearing seals 4024 and an optional circumferential oil reservoir 4025 fed by oil passage 4026. The oil reservoir may be in the form of an oil saturated compressible material or wick. The tribological agent may be any substance of any composition, including conventional oils, heavy oil, grease, and may be gravity or pressure fed. In this embodiment the oil passage 4026 continues within hydrofoil 4010 to its extremity to terminate in another circumferal reservoir 4027 and seal 4028, both having the approximate form of the cross-section of 4010. The seal is retained by a removable flange or fin 4029. The reservoir 4027 may comprise a lubricant soaked material or wick, with its constituent parts such as fibers so arranged as to absorb oil/ lubricant from the surface of foil extension 4011 during extensible motion, and to give

up oil onto the surface of 4011 during its retraction. Any suitable tribological material, including oil, etc. may be used. Either the inner surface of 4010, or the outer surface of 4011, or both, may house a series of inserts 4030 which act as bearing surfaces, especially useful if the hydrofoil elements are made of relatively soft material(s). For example, the inserts may be of ceramic material(s). Elements 4010 and 4011 may have locating pins 4031 and/or locating grooves 4032 to define the limit of extension of 4011 relative to 4010. Although not shown, a broadly similar system of inserts can be used to reinforce the bearing surfaces of disc 4013 and keel element 4005. 4038 is interior volume of 4005.

Essentially the action of 4011 relative to 4010 is telescopic, as illustrated herein with a single telescopic element. In fact, more than one extensible/retractable (i.e., telescopic) element may be used in association with one hydrofoil, on a similar principle and constructional details disclosed herein with reference to hydrofoils may also be applied to the extensible/retractable post(s) linking keel element(s) to hull, each post having either a single or multiple telescopic elements.

The hydrofoils may be deployed on the keel(s) in any way. In the examples that follow constant size hydrofoils are generally shown, but where appropriate extensible/retractable hydrofoils can be used. Hydrofoils herein are generally shown by way of example having symmetrical cross-section and linear longitudinal section. In fact, they may have any cross-section, including wing-shaped, and any longitudinal section, including curved, in any longitudinal plane. If the longitudinal section is of constant radius curvature, or if the cross-section is of any type, then the hydrofoil may have extensible/retractable elements. Likewise the posts may have any appropriate cross-section and longitudinal section, whether extensible/retractable or not. Herein the extensible/retractable hydrofoil(s) and post(s) have been described as having telescopic action, but in fact any suitable system of extension/retraction may be employed. The posts may be at any appropriate angle to the water line, as may be the

hydrofoils. In some embodiments the hydrofoils may be so angled that they are partially above the water line during periods of operation, at such moments giving reduced lift, in the manner of some protected water hydrofoil craft (for example, as currently used in Sydney harbor, Australia). Mostly a given hydrofoil loses lift efficiency of within a certain depth below the water surface, so generally it is intended that hydrofoils operate continuously submerged, preferably below the depth within which lift performance becomes significantly impaired.

The range of hydrofoil/keel element/post combinations is virtually limitless. By way of example, four layouts are shown in Figures 444 through 447, in which each figure (a) is a schematic side elevation, (b) a schematic front elevation, and (c) a schematic plan of the keel element/hydrofoil configuration. Each individual hydrofoil can be free-standing, or be supported by two struts 4036 as in Figure 448, or by a single strut as shown in Figure 449, or by a multiplicity of struts (not illustrated). Figure 449 shows an adjustable pitch hydrofoil supported by struts or land straps, with the axis of pitch adjustment shown at 4033. The struts may have any appropriate cross- and longitudinal sections which may optionally provide positive or negative lift. Struts themselves may have adjustable pitch, as shown in Figure 449, to provide variable lift, and may be considered as a form of hydrofoil. Hydrofoils may have flaps 4034 to provide variable positive or negative or rapid braking (similar to flaps on aircraft wings) as shown in Figure 450. Figure 450 also shows a pivotable flap 4035, axis of rotation 4036 here in a keel element to provide variable side thrust on the keel. Any kind of hydrofoil elements may have extensible, retractable components, including rudders, as shown schematically in Figure 437 wherein extensible component 4037 in retractable within ruder 4007.

An extensible/retractable post will cause the keel element to be retracted toward the hull up to a final position either clear of the hull, or up against the hull, or in some embodiments to partly or wholly within a recess in the underside of the hull.

Figures 451 and 452 show schematically by way of example such a keel element 4005 and hull 4001 relationship, wherein Figure 451 is an elevational view showing the keel element 4005 in its uppermost, partially recessed position, while Figure 452 is a section through A taken when the keel element is in a lowered position, a small rudder is shown at 4007. No conventional hydrofoil 4006 is indicated, because in certain applications the keel element may itself have a hydrofoil or lift effect, as indicated here. The longitudinal section of the more or less horizontal parts 4037 of the keel element are of wing-like cross-section, so provide lift. The turned down parts 4038 of the keel element restrict the sideways flow of water from underneath the keel element and so contribute to additional support. In figures 451 and 452, section A is taken at the widest part of the keel element, which narrows before and rearward of section line A. Figure 453 shows in schematic plan view a different type of keel element which is integral with main hydrofoils to produce a new type of composite lifting surface 4040, fitted with flaps 4034, conventional rear hydrofoils 4006 pivotable about 4033 and rudder 4007 pivotable about 4007a. The upper or back fin 4039 acts as a directional stabilizer and may be recessed as shown in Figures 451, 452, or not. A section taken through B will be very similar to Figure 452, but proportionately expanded width. The forms of Figures 451 to 453 may not be the most efficient in terms of drag and lift effect, but they could be simply and cheaply constructed, and so could be suitable for small pleasure craft with high power to weigh ratios. The keel element of Figures 451, 452 may be used in conjunction with convention main hydrofoils or rear hydrofoils (as for example in Figures 426, 444 through 450), especially if not required to be retractable into the hull recess to the degree shown. (It need not be retractable at all.) The keel element(s) of Figures 426 to 451 may be used in conjunction with any drive and/or power unit system (as for example in Figures 431 to 437). By way of example, Figure 454 shows schematically another layout, being an elevational view of an enlargement of within the square indicated in Figure 451. Here post 4004 is not separate from but integral with keel element including surfaces/parts 4037, 4038 and back-fin 4039. This same integral post/keel element construction can be used in any application, including

those illustrated in this disclosure. An engine 4044 is located below a waterjet drive 4043 supplied by water via passage 4042 from entry at 4041. A stub keel 4045 is provided to take loads (and transfer them to post) in the event of grounding. A section through C will be broadly cross-shaped, unlike the broadly inverted "T" shape of Figure 452. If the water jet drive need not be located so high, then the positions of drive and engine could be broadly transposed, thus avoiding the need to cross air supply (not shown) with propulsion water flow. Figure 455 shows, in a manner similar to Figure 454, a layout of a more conventional post/keel element/hydrofoil combination, where the engine is higher than the jet drive and is located within a sort of pod 4050 on or part of the keel element 4005. When the post/keel element combination is retracted to its furthest position, a lip on the pod or projection 4050 presses against a continuous seal 4049, which may be of any appropriate material, including an inflatable continuous tube. Should the engine and/or drive require maintenance, the space 4052 between the pod 4050 and the depression in the hull necessary to accommodate it, 4046 is pumped clear of water, after which the housing which comprises 4046 is removed, permitting access to the top of the pod(and therefore engine and drive) from within the hull, space 4047. By removing the top of the pod 4050 access is obtainable to any of the volume 4048 of the keel element. The need to provide a lip on 4005 and/or 4050 may be used to create another surface 4051 having modest hydrofoil effect.

Figures 457, 458 and 459 show schematically in half plan view, elevation and sectional profile of numbered section line a fast marine craft with twin post/keel assembly/hydrofoil assemblies, retractable at an angle shown in Figure 459. Each post/ keel assembly has its own drive system and engine (not shown). The vessel has a third drive assembly and engine indicated at 4053 and a third rudder 4054 to propel.control the vessel with hull in the water at low speed. At high speed the hull lifts out of the water and is steered by rudders 4007. Line of possible mast is shown at 4055. To enable the assemblies to extend sufficiently, a housing 4056 accommodates the posts when retracted. A central forward fixed keel 4057 is provided to improve hull-in-the-

water directional stability, and to probably ground first (thereby protecting the keel assemblies when in the retracted position). A feature of the hull is the extra depth and buoyancy 4058 on each side, over the keels when retracted, and the concave hull form between. The keel assemblies are shown butting up against the hull when retracted, but they could be partially recessed as shown previously. This hull shape has a tendency to trap some of the bow wave under it and so lift the vessel, enabling it to plane move.

## 5. Compounding

This section deals with ways of using the heat energy of the engine exhaust gas, other than by turbocharging.

Exhaust gas can be used to vaporize water, i.e., produce steam, especially in marine applications where water is readily available. Systems are known, wherein the steam is put to work in a second engine (the reciprocating engine portion and the steam engine portion together comprising a compound engine). In marine engines, gas consisting of exhaust only, or steam only, or a mixture of both can be used either to create additional propulsive thrust and/or to create a relatively incompressible bulk at an appropriate underwater location at or near a drive system, such as a propeller or water jet.

In Figures 460, 461, a base embodiment of the thrust embodiment is shown, wherein Figure 461 is a section along "A" in Figure 460, 4500 is the direction of normal forward motion through the water 4501, thrust being generated in direction 4502, flow of hot exhaust gas shown at 4503. A "J" shaped exhaust pipe 4504 is shown in a preferred embodiment having a progressively varying cross-section (a constant cross-section would also be functional). It is made of any suitable material, but in the preferred embodiment is of ceramic having heat insulating quality. It is mounted below

the water line 4505 to the side of a hull or engine housing 4506 by means of a casing 4507 preferably of ductile material. An interlayer 4511 of material such as ceramic wood or other compressible substance is shown between casing and pipe and casing and hull/housing, although in some embodiments this could be omitted. The casing is preferably of metal or fiberglass, as is the hull/housing. The objective of the mounting method shown is to have the casing act as shock absorber if striking objects or fish, and so protect the integrity of the pipe 4504. Both pipe and casing is cast round a ceramic pipe and optional interlayer. Otherwise the casing and/or pipe can be each of multiple assembled portions (not shown).

In operation, the forward motion of the marine vehicle causes, to significant degree by ram effect, water to pass through a series of aligned apertures 4508 in both casing and pipe and enter the exhaust gas flow is thin jet or droplet 4509 form. The apertures are shown longer than they are in practice likely to be, for the sake of illustrative clarity. Preferably the aligned apertures are of larger cross-section on the water/casing side than the exhaust gas/pipe side, so that the ram effect can accelerate the water through the apertures. To prevent debris clogging the apertures, an optional mesh screen is shown at 4510. Once in the exhaust gas flow, the jets and/ or droplets will be broken up into smaller units by the kinetic energy and turbulence of the exhaust gas, and these smaller jets/droplets will tend to boil with exposure to the hot exhaust gas, creating steam within the gas exiting the pipe. The additional steam will cause the gas to exit at a greater velocity than it would if it were only exhaust gas. The gas exit velocity under a constant state cruising condition will be at least partly determined by the size of the apertures 4508, which determine how much water is admitted, the temperature (i.e., heat content) of the exhaust gas, and the cross-sectional area of the pipe exit. Optional turbulence inducing devices 4511 may be placed within the pipe, of a configuration similar to the filamentary material described earlier herein. The turbulence will improve mixing of droplets with gas, and in addition the hot surfaces of devices 4511 will cause any water deposited on them to boil away.

Obviously, it will be desirable to size the apertures to admit just sufficient water at optimum cruising speed to lower the exhaust and water vapor mixture temperature to a little over 100°C. In other embodiments, for example where for environmental reasons mean exit temperatures should be lower, more water could be admitted via the apertures, with thrust being generated by a mixture of gas and water accelerated by gas. If only a little more water is added, nearly all the water would initially be converted to steam, and at the exit some of the steam would have condensed out or be converted to vapor, i.e., be below 100°C. The entry of water into the gas flow, and the water's progression through various phases or stages, can be regulated to clean the exhaust gases and control emissions into the water and/or atmosphere.

In the disclosure, marine or other exhaust gas/steam/water vapor thrust or cleansing devices are generally described as being associated with the reciprocating IC engines of the disclosure. In fact, any such devices described previously or subsequently herein can be used in combination with any engine, or water may be heated and/or expanded by any appropriate means.

The exhaust gas need not be mixed directly with water, but may generate steam and/or water vapor by means of heat exchangers. In such case, the exhaust gas may be discharged separately from the water products,either into the water or into the air. In marine applications, the water could be pre-heated by being passed through heat exchangers located in a plenum containing compressed charges (thereby cooling the charge) and thereafter be admitted to the exhaust gas. In certain applications such water could be passed under pressure through the heat exchangers and to the exhaust gas entry apertures, and its quantity and flow be so regulated as to be superheated by the compressed charge, i.e., be at a temperature significantly higher than 100°C. On discharge into the exhaust gas flow, some of the superheated water would immediately vaporize, while the remainder would use less exhaust gas heat energy to vaporize or

become steam. In other words, pre-heated water means a given energy-content exhaust gas can create more steam/water for cleansing or thrust purposes.

By way of example, such a system is shown schematically in Figure 462, wherein 4513 shows an engine pod mounted on a keel (not shown) having an engine driving two contra-rotating propellers 4514. An aperture system 4508 at the front of the pod leads water via passages 4515 to heat exchangers 4516 located in a plenum 4517 through which compressed charge flows, and thence to the exhaust gas flow 4503 at a point just inside each exhaust pipe 4504, so shaped to direct the gas thrust clear of the propellers. In an alternative embodiment not illustrated, the water after being heated by the compressed charge would flow to heat exchangers located in a hot exhaust plenum, be converted to vapor/steam on being released into the "exhaust" pipe. The water need in fact carry no exhaust gas, which could be directed from the plenum to the outside air via passages in the keel. In another embodiment not illustrated, wherein the engine in the pod has virtually no forced aspiration, water goes directly from an aperture system to a heat exchanger located in the exhaust gas plenum. Where water is pre-heated before contact with exhaust gas, such pre-heating could be from any source of energy other than charge air, including electric heaters.

Rather than have a separate gas discharge pipe (as 4504 in Figures 460 through 462), the gas(es) can be discharged along the center of rotation of a drive system. Figure 463 schematically illustrates this principle in its simplest form. The interior of the hollow propeller shaft 4518 carries hot exhaust gas flow 4503. An aperture system 4508 located where the shaft bells out to support the propeller blades 4514. The conversion of water to vapor or steam at 4509 creates an accelerated, initially more-or-less horizontal column of gas, diameter indicated at 4519 discharging in direction 4502. In addition to providing some thrust, this "column" of gas improves propeller performance in two ways: it prevents the water collapsing in on itself behind the core of the propeller, and it permits a larger ratio of hub to overall propeller

diameters, thereby enabling the inefficient restricted passages between blade roots to be reduced, as a percentage of total propeller water clearance space. Alternatively, propellers having a greater number of blades for a given swept area may be constructed. Figure 463 shows an unshrouded propeller--the above principles may equally be applied to shrouded propellers (not shown). The principle of gas flow through the center of drive rotation may apply to any gas, including unadulterated exhaust gas, from any source, in the preceding and subsequent disclosures. The hollow propeller shaft may either be of insulating material and/or have a lining of insulating material applied to the exterior or interior of the tubular shaft. which may be of constant or progressively variable cross-sectional diameter. Water at any temperature may be delivered by any means to the water/exhaust gas mixing point, or if superheated and/or under pressure to a pressure release/discharge point, including by capillary tubes or passages within propeller shaft wall thickness.

Gas may alternatively or additionally be used to reduce friction between water and propeller or impeller blade surfaces. In the schematic embodiment of Figure 464, gas flows 4503 along a hollow propeller shaft, through hollow blades, shown dotted at 4520, and discharged to the water by a series of closely spaced apertures 4521, or a narrow permeable strip (not shown), located at or near the leading edge of each blade. The forward motion of the vehicle will cause a laminar flow of gas 4523 between water and blade surface. A similar flow can be created over propeller hub nacelle 4522 by means of a series of closely spaced apertures, or a narrow permeable strip, located at 4524 ahead of blade roots. The features of Figures 463 and 464 can be combined, optionally by having two differing gases or gas systems.

Especially but not exclusively in shrouded drive system, the presence of a coaxial gas system will permit a propeller or impeller hub of progressively varying diameter, to match the acceleration of the water across the blades. Figure 465 illustrates this principle schematically and shows a shroud 4525 supported by stator

4526 anchored to a drive shaft housing 4527. Impeller blades 4528 rooted to a belled shaft end 4529 in operation accelerate water in the opposite direction to forward motion 4500. Water enters the system through cross-sectional area represented at "a" at a speed roughly equivalent to forward motion; if it is accelerated the water flow will occupy less cross-sectional area represented at "h." Any gas system flows through the hollow shaft and belled end 4529 to both provide thrust and provide a core to support accelerated water tube at 4502. In the arrangement of Figure 465, the features of Figure 464 may also be incorporated--gas may be directed through the leading edges of shroud 4525, stators, impellers and the trailing edge of housing 4527.

An alternative method of reducing the friction of water flows across impeller/stator blades, shrouds, etc. by introducing a very thin film of gas between water and component surfaces is by using heat, preferably at the leading edges of such components. For example, in Figure 464, where there are now a series of closely spaced apertures at 4521, there would alternatively be a heated area at a similar location. The heat would cause just a few of the molecules of water passing over the area to vaporize, sufficient to produce a thin gas film, which would adhere to the surfaces under optimum conditions due to laminar flow. The local heat can be provided directly by the exhaust gas or indirectly by liquid circulating through a heat exchanger placed in the exhaust gas flow. Alternatively, it can be provided by any other appropriate means, including electricity.

If gas is used to reduce friction, then it can be directed along the appropriate surface by exiting from a continuous slit. Figure 466 shows this schematically, being a cross-section through a propeller or impeller blade which is hollow and constructed with the surfaces in two portions linked by internal structural support. The laminar flow of gas past the main body of the surface is indicated at 4530. Obviously, if the gas is heated, various insulating material can be disposed inside the blade to give varying surface temperatures.

Currently available water jet drives, such as manufactured by Kawema in Sweden, have fixed inlet and outlet openings, the latter being about half the cross-sectional area of the former to accommodate the acceleration of water achieved by the impellers. To reverse the marine craft, the water flow through the drive remains unchanged but deflector plates are moved into position rearward of the outlet nozzle to deflect the thrust through 130° to 180° from normal direction. The cumbersome and weighty deflector plates and actuating mechanism when in the retracted position for normal forward motion causes substantial drag. Using gas not only to provide thrust but also bulk within a drive permits the elimination of reverse deflector mechanisms. Instead, the rotational direction of the impellers is reversed, water is sucked into the normal outlet aperture and driven out through the normal inlet aperture, water acceleration and reduction of cross-sectional exit area being achieved by reversal of the gas flow within the drive.

Figure 467 shows schematically a jet driving having shroud 4525, fore stators 4526, rear stators 4537, fore impeller 4531 turning in one direction and mounted on bell ended propeller shaft 4538, aft impeller 4528 turning in the other direction mounted on bell ended propeller shaft 4539 mounted coaxially within shaft 4538, bearings and/or gland systems at 4532, propeller shaft housing 4527. Gas flows at 4503 with the shafts to exit the shaft bell ends via holes 4536. In normal operation, the vehicle moves in direction 4500. When travel in the opposite direction is desired, the direction of both impellers is reversed. Previously, in forward motion the gas has travelled through the drive in direction 4533 and thrust has been created in 4503, with reversal, the gas now flows in direction 4534 and thrust is created in direction 4535. It will be noted that in normal operation impeller 4531 drives non-aerated water, while impeller 4528 drives a mixture of non-aerated water, aerated water and gas. Because gas has a tendency to rise to the surface, arrangements can be made to discharge more gas below the center of rotation than above it, preferably by having uneven distribution of holes within stators, stator hubs or shrouds. Any kind of gas can be used. including

that containing steam or water vapor. Figure 468 shows a drive herein more gas issues below the center of rotation 4543 than above, the water surface being in direction 4542. A bell shaped propeller shaft end and/or hub 4540 has mounted on it impeller/propeller blades 4541 and continues holes 4536 for gas exit. Exhaust gas flows 4503 past an eccentrically and fixedly mounted water delivery tube 4544 within the shaft whose delivery aperture is located near the lowest point of the volume within the hub, causing a majority of the water vapor/steam to exit the hub at 4545, below the axis of rotation.

# TABLE 1 – COMPARATIVE PERFORMANCE

| Engine no: | #1 | #2 | #3 | #4 |
|---|---|---|---|---|
| Engine type | Volvo TD 100B 1982 4 stroke diesel | Napier Nomad 1953 2 stroke diesel | CAICE fixed CR 1984 2 stroke diesel/multifuel | CAICE variable CR 1986 2 stroke diesel/multifuel |
| charge boost × compression ratio | est. 2.4:1 × 17:1 | 6.3 to 8.3:1 × 8:1 | 3:1 × 20:1 | 6:1 × ① 20 to 100:1 |
| no cylinders × comb. cham. | str. 6×6 | flat 12×12 | str. 5×10 | packed 62·64 |
| swept volume in³ | 585 | 2508 | 527.5 | 4431 |
| est. oa dimensions, ft | 2.5×3.7×4.0 | 3.3×4.7×8.5 | 3.0×.85×3.9 | 3.0×3.2×5.7 |
| est bulk incl. ancill, ft³ | 28 | 92 | 10 | 55 |
| total weight, lb | 2029 | 3579 | 320 | 1980 |
| weight per ft³, lb | 72.5 | 38.9 | 32 | 36 |
| Torque at 17 rps (1020 rpm) ft-lb | 760 | est. 7300 | 3092 | 42,619 |
| Torque at 105 rps (6,300 rpm) ft-lb | – | – | 3455 | 47,633 |
| HP at 17 rps (1,020 rpm) | est. 100 | est. 1300 | 601 | 8,276 |
| HP at 35 rps (2,100 rpm) | 284 | 3066 (Max) | 1308 | 18,034 |
| HP at 105 rps (6,300 rpm) | – | – | 4146 (Max) | 57,131 |
| HP at 259 rps (15,540 rpm) | – | – | – | 163,289 (Max) |
| HP/in³ @ 17 rps (1020 rpm) | .171 | .518 | 1.139 | 1.868 |
| HP/in³ @ 35 rps (2,100 rpm) | .485 | 1.215 | 2.48 | 4.07 |
| HP/in³ @ 105 rps (6300 rpm) | – | | 7.86 | 12.89 |
| Max. HP/in³ (#1 - 300 @ 37 rps) | .513 | 1.215 | 7.86 | 37.93 |
| Improvement factor over #1 | – | 2.37 | 15.3 | 76.0 |
| Improvement factor over #2 | .42 | – | 6.47 | 31.3 |
| Max. HP per lb weight | .148 | .851 | 13.0 | 85.0 |
| Improvement factor over #1 | – | 5.8 | 87.5 | 574.3 |
| Improvement factor over #2 | .174 | – | 15.2 | 99.9 |
| Max. HP per ft³ bulk | 10.7 | 33.1 | 414.6 | 3059.6 |
| Improvement factor over #1 | – | 3.09 | 38.7 | 286.0 |
| Improvement factor over #2 | .323 | – | 12.5 | 92.4 |

① Engine #4 operates at 20:1 Cr up to piston take-off speed of ± 120 rps (7,200 rpm) reaching maximum 100:1 Cr at 259 rps (15,540 rpm)

## TABLE 2 : VARIATION OF PARAMETERS WITH COMBUSTION CHAMBER GEOMETRY

| Engine Type (see Fig 353 [363]) | A | B | C | D |
|---|---|---|---|---|
| Volume: Units$^3$ | 50·3 | 150·8 | 251·3 | 502·6 |
| Piston Speed (ave) at 100 rps: Units ps | 800 | 800 | 800 | 1600 |
| Piston speed per unit volume: ratio | 15·9 | 5·3 | 3·2 | 3·2 |
| Stroke per CR of 16:1: ratio | ·25 | ·25 | ·25 | ·5 |
| Stroke per unit volume: ratio | ·079 | ·027 | ·016 | ·016 |
| Chamber surface area (exc. piston): units$^2$ | 62·9 | 138·2 | 213·6 | 364·4 |
| Surface area per unit$^3$ volume: units$^2$ | 1·26 | ·92 | ·85 | ·73 |
| Seal lineage: units | 23·9 | 37·7 | 62·8 | 62·8 |
| Seal lineage per unit$^3$ volume: unit | ·475 | ·25 | ·25 | ·125 |

# EFFECT OF REDUCTION OF RECIPROCATING MASS

The formula relating to the stresses induced by the reciprocating masses is:

Force = Mass x Acceleration

$$= 2.84 \times 10^{-5} \times M \times n^2 \times r \times (\cos\theta + r/l \cos 2\theta) \text{ lb},$$

where M is the weight, n the rpm; r the crank radius in inches, l the length of the connecting rod/link in inches.

For a given engine the Inertia Force Factor $(\cos\theta + r/l \cos 2\theta)$ is constant for a given crank angle $\theta$. For example if $l/r$ is 3.75 (ie $r/l$ is 0.267), the I.F.F. values will range from 1.266 at 0° to -0.734 at 180°. The crank radius r is constant, as is the conversion factor $2.84 \times 10^{-5}$. Considering only the variables M and n, it will be seen that if reducing M, the forces will remain constant if n is increased. A four-fold reduction in M will permit twice the engine speed for a given stress limit. See also Note 6: "Piston Take-off and Variable Compression Ratio."

# THERMAL CONDUCTIVITY

Assuming theoretically perfect insulation about the crankcase and exhaust processing volumes (considering the basic embodiment of Fig. 189), it will be seen that the only constant heat loss from the combustion chambers, apart from that carried away by the exhaust gases, will be that through the head to the crankcase. Of course this heat will not in reality be lost, since it will heat the charge and be carried back to the engine cycle. To form a very rough estimate of what this heat transfer will be, it will be assumed that the head has an average thickness of 1.4 inches, that questions of valve design are ignored, and that the head material is aluminum oxide $Al_2O_3$.

The thermal conductivity of $Al_2O_3$ is given as approximately .005 BTu per square foot per °F temperature differential per second per inch depth of material. If the combustion chamber has both stroke and bore of 4", then the internal head area will be 12.6 in² or 0.0875 ft². Assuming that average temperature differential will be 1500 °F, then

$$T_k = \frac{.005 \times .0875 \times 1500}{1.4} = 0.469 \text{ BTu per second.}$$

Assuming that 5.0 BTu of fuel energy is provided at each firing and ignoring energy converted to mechanical work, then for two stroke engines percentage of total energy conducted away:

At
$$4 \text{ rps } (240 \text{ rpm}) = \frac{.469}{4 \times 5.0} = 2.34\ \%$$
$$10 \text{ rps } (600 \text{ rpm}) = \frac{.469}{10 \times 5.0} = 0.94\ \%$$
$$35 \text{ rps } (2100 \text{ rpm}) = \frac{.469}{35 \times 5.0} = 0.27\ \%$$
$$105 \text{ rps } (6{,}300 \text{ rpm}) = \frac{.469}{105 \times 5.0} = 0.09\ \%$$

NOTE 3



## COMBUSTION CHAMBER BLOW-BY

It is assumed that piston/cylinder tribology will be based on gas bearing design principles, with particularly small clearances between moving surfaces of the order of 2 microns (0.000,0787 in. Although it is not anticipated that blow-by will be significant between head and tensile member (see constructional descriptions relating to fuel delivery, bearing support by water, etc), to allow for wear and mis-alignment of valves a clearance gap of 2 microns round a rod of .75 in dia attached to a piston of 4 in. dia has been allowed for. Total blow-by area is therefore:

$$A = \pi \times 2.0000787^2 - \pi \times 2^2 + \pi \times .7500787^2 - \pi \times .75^2 = .001175 \text{ in}^2$$
$$= .000,0082 \text{ or } 8.2 \times 10^{-6} \text{ ft}^2.$$

Theoretical maximum gas velocity (speed of sound) is reached when the pressure differential across the blow-by gas reservoirs is two-fold or greater. In a given combustion chamber, piston blow by losses meet this condition about 37% of cycle time (maximum is 1/2, since pressure can only exceed that in opposed combustion chamber on other side of piston for half total operating time). To allow for more continuous head/rod losses, the condition is assumed fulfilled 40% of cycle time. The formula for maximum gas blow-by losses is given by:

$$W = \frac{71.6 \times A \times P_1}{\sqrt{T_1}} \quad \text{when } P_1 > P_2 \times \pm 2 \text{ (see Postscript ①)}$$

W is the airflow in lbs/sec, T is temperature °R, A is area in ft², $P_1$ is combustion chamber pressure in lb/in².

A mean working gas temperature of 4,600 °R and pressure of 950 lb/in² are assumed during blow-by loss condition. Therefore

$$W = \frac{71.6 \times .0000082 \times 950 \times .4}{\sqrt{4,600}} = .00323 \text{ lb/sec.}$$

..... continued over

Assume each firing consumes .005 lb of charge (see note 5), then the percentage of charge lost through blow-by is at:

$$4 \text{ rps } (240 \text{ rpm}) = \frac{.00323}{4 \times .005} = 16.15\%$$

$$10 \text{ rps } (600 \text{ rpm}) = \frac{.00323}{10 \times .005} = 6.46\%$$

$$35 \text{ rps } (2100 \text{ rpm}) = \frac{.00323}{35 \times .005} = 1.85\%$$

$$105 \text{ rps } (6300 \text{ rpm}) = \frac{.00323}{105 \times .005} = 0.62\%$$

$$259 \text{ rps } (15{,}540 \text{ rpm}) = \frac{.00323}{259 \times .005} = 0.25\%$$

The above are maximum percentages. In reality percentage losses will be significantly lower. If piston rings are fitted. losses will be greatly reduced. (The calculations are for an unguarded two micron piston/cylinder clearance gap). About half the loss will be exhaust gas at high temperature and pressure to the exhaust syste[m] where its energy is recoverable. Nearly all the remaining loss will be air and fuel to the exhaust system, useable if afterburning is employed. (See Appendix C. Some unaided afterburning will occur in the exhaust reservo[ir] if both fuel and oxygen are present, especially at high temperature and pressure).

Postscript ①

The given formula was derived from (where $p$ = critical pressure):

$$\text{Max. velocity} = 49.1\sqrt{T} \text{ ft/sec}; \quad \frac{1}{\rho} = \frac{53.3 \times T}{144 \times p}, \quad \therefore \rho = \frac{144p}{53.3T}; \quad p = .54$$

$$W = \rho A V = A(49.1\sqrt{T}) \times \frac{144}{53.3T} = \frac{133\,A\,p}{\sqrt{T}} = \frac{133\,A \times .54P_1}{\sqrt{T}}$$

$$= \frac{71.6 \times A \times P_1}{\sqrt{T}}$$

# CRANK GAS BEARING REQUIREMENT

It is assumed each bearing has an area which is that fraction of piston crown area as the portion of total piston loads it has to carry, and that a variable system always provides bearing pressure corresponding to combustion chamber pressure at that time. Assuming also a flexible connection at the head end of the tensile link, then only a main crank and a main crank throw bearing are required. If each bearing has effective area of 12·6 in² and may be 5¼" long, then each will have a diameter of 2·4". If gas escape can only be at bearing ends, and if clearance gap is two microns, then total escape area is

$$(\pi \times 1{\cdot}2000787^2 - \pi \times 1{\cdot}2^2) \times 2 \times 2 = {\cdot}0023736 \text{ in}^2 = {\cdot}0000165 \text{ ft}^2$$

If combustion chamber pressures average 500 lb/in² over the complete two stroke cycle and mean crankcase temperatures are 900°R (see note 5), then using the formula given in note 3:

$$W = \frac{71{\cdot}6 \times {\cdot}0000165 \times 500}{\sqrt{900}} = {\cdot}01969 \text{ lb/sec}$$

Assuming that the gas bearing is only effective above 10 rps (600 rpm) and that each firing consumes ·005 lb of charge, charge and bearing gases having equivalent densities, then bearing gas requirement as a proportion of engine gas requirement will be:

$$\text{At } 10 \text{ rps } (600 \text{ rpm}) = \frac{{\cdot}01969}{10 \times {\cdot}005} = 39{\cdot}4\,\%$$

$$\text{At } 35 \text{ rps } (2100 \text{ rpm}) = \frac{{\cdot}01969}{35 \times {\cdot}005} = 11{\cdot}3\,\%$$

$$\text{At } 105 \text{ rps } (6300 \text{ rpm}) = \frac{{\cdot}01969}{105 \times {\cdot}005} = 3{\cdot}7\,\%$$

$$\text{At } 259 \text{ rps } (15{,}540 \text{ rpm}) = \frac{{\cdot}01969}{259 \times {\cdot}005} = 1{\cdot}52\,\%$$

This requirement will be greatly reduced if the ends of each

bearing are closed by an oil film. The higher the bearing gas temperature the more reduced the requirement. If bearing support gases are to be supplied from the combustion chamber, they can be supplied by increased charge boost, to compensate for the controlled bleed-off via passages to crank bearings (such bleed-off when occurring during compression being optionally part of an of an 'overlong' expansion engine design). Off course, if the bearing gas is air, no real bleed-off "loss" will occur in the basic two stroke embodiment of Fig. 144, since all high pressure air loss will go into the charge reservoir contained by the crankcase.

It is clear from the above that bearing gas requirents could be relatively substantial at low engine speeds. For engines required to run at a wide speed range, hybrid bearings (such as the gas/oil bearing described) are probably required.

NOTE 5 

# TEMPERATURE, PRESSURE, WORK, EFFICIENCY

## FIXED COMPRESSION ENGINES

Theoretical calculations have been made for a single combustio chamber of engine A and engine B (described below), using the ideal gas and adiabatic compression and expansion laws, as applied to the processing of one pound of air. Where appropriate the figures have been applied to actual charge of the combustion chambers of engines A & B, which are effectively identical excep for turbo-boosted induction pressure.

Hottels' tables and graphs have not been used to allow for disassociatio effects, mainly because the regions under consideration here lie outsi the standard editions of these graphs. The elevated pressures will tend to inhibit disassociation, but some has been allowed for by subtracting a percentage of total fuel energy from the combustion process and adding it to the exhaust gases. Because these are intended as high speed engines, the Otto cycle has been used in the calculations, where pressures are in lb/in², temperatures °

No allowance has been made for dilution of the charge by residual exhau gases. Either the capacity can be increased proportionally, or exhaus gas bleed-off pockets (see constructional details) can be providec [illegible] increasing the capacity - or more charge blow-throu can be provided. Similarly, no allowance for tensile member radi [illegible], since the bore dimension can be adjusted to provide a constant volume. Comparing engines A & B, increasing tensile [illegible] diameter from 1/2" to 3/4", increases the bore by ·038". Th calculations are based on air handled, not constructural arrangemen [illegible] process. Values for $C_v$, $C_p$ and k used are based on estimated mean temperatures during that process..

| Engine A | | Engine B |
|---|---|---|
| | **Combustion Chamber Details** | |
| 52·75 | Capacity, $in^3$ | 52·75 |
| 2·64 | Clearance Space, $in^3$ | 2·64 |
| 50·11 | Swept volume, $in^3$ | 50·11 |
| 4·0 | Stroke, in | 4·0 |
| 12·53 | Piston crown area ($\pi r^2$), $in^2$ | 12·53 |
| 0·20 | Tensile rod sectional area, $in^2$ | 0·44 |
| 0·5 | Tensile rod diameter, in | 0·75 |
| 4·026 | Cylinder diameter (bore), in | 4·064 |
| 44·1 | Charge boost pressure, $lb/in^2$ | 88·2 |
| 3:1 | Charge boost ratio | 6:1 |
| 14·7 | Ambient pressure $P_1$, $lb/in^2$ | 14·7 |
| 500 | Ambient temperature $T_1$, °R | 500 |
| 12·2 | Volume 1 lb ambient air, $ft^3$ | 12·2 |

## Turbo Charge

| Engine A | | Engine B | |
|---|---|---|---|
| $V_3$ | 5·73 | 3·46 | $V_4$ |
| $T_3$ | 683° | 825° | $T_4$ |
| $P_3$ | 44·1 | 88·2 | $P_4$ |

Given $P_3, P_4$, then $\frac{T_3}{T_1} = \left(\frac{P_3}{P_1}\right)^{\frac{k-1}{k}}$

$T_3 = T_1 \times (P_3/P_1)^{k-1}$ (adiabatic compr.)

$T_3 = 500 \times 3^{\frac{.396}{1.396}} = 683$

$T_4 = 500 \times 6^{\frac{.388}{1.383}} = 825$

$P_3 V_3 = M \times 53.3 \times T_3$ (gas law)

where $P$ is in $lb/ft^2$, $M$ is mass of gas.

$V_3 = \frac{53.3 \times 1 \times 683}{44.1 \times 144} = 5.73 \ ft^2/lb$

$V_4 = \frac{53.3 \times 1 \times 825}{88.2 \times 144} = 3.46$

....continued over

| Engine A | | Engine B |
|---|---|---|

**Work of turbocharging:**

$W_3$ 26,674 ft-lb

$$W_3 = \frac{P_3 V_3 - P_1 V_1}{k-1} \text{ ft-lb}$$

46,701 ft-lb $W_4$

$$W_3 = \frac{44.1 \times 144 \times 5.73 - 14.7 \times 144 \times 12.2}{.376}$$

$$W_4 = \frac{88.2 \times 144 \times 3.46 - 14.7 \times 144 \times 12.2}{.308}$$

## Compression

Both volumes are compressed 20:1. Therefore $V_5 = .3218$, $V_6 = .2079$

$$T_5 = T_3 \times (V_5/V_3)^{k-1} \text{ (adiabatic comp.)}$$

| Engine A | | Engine B | |
|---|---|---|---|
| $V_5$ | .2865 | .173 | $V_6$ |
| $T_5$ | 1931° | 2251° | $T_6$ |
| $P_5$ | 2495 | 4816 | $P_6$ |

$$T_5 = 683 \times 20^{.347} = 1931$$

$$T_6 = 825 \times 20^{.335} = 2251$$

$$P_5 = \frac{53.3 \times 1 \times 1931}{144 \times .2865} = 2495$$

$$P_6 = \frac{53.3 \times 1 \times 2251}{144 \times .173} = 4816$$

Work of compression:

$W_5$ 191,775 ft-lb

$$W_5 = \frac{2495 \times 144 \times .2865 - 44.1 \times 144 \times 5.73}{.347}$$

226,959 ft-lb $W_6$

$$W_6 = \frac{4816 \times 144 \times .173 - 88.2 \times 144 \times 3.46}{.335}$$

## Combustion

Assume heat content of fuel is 19,800 BTu/lb, that air/fuel ratio is 19:1, and that 3% of fuel energy is taken out of combustion process and returned to exhaust gas, due to effects of disassociation.

...continued over

Note 5 continued



| Engine A | | Combustion continued: | Engine B | |
|---|---|---|---|---|

Energy input, Q, in BTu:

$Q = M \times C_v \times (T_7 - T_5)$

$\therefore T_7 = \frac{Q}{M \times C_v} + T_5$

$Q = 19{,}800 \div 19 \times .97 = 1010.84$ BTu

| | | | | |
|---|---|---|---|---|
| $V_7$ | .2865 | $T_7 = \frac{1010.84}{1 \times .234} + 1931 = 6250$ | .173 | $V_8$ |
| $T_7$ | 6250° | $T_8 = \frac{1010.84}{1 \times .235} + 2251 = 6552$ | 6552° | $T_8$ |
| $P_7$ | 8075 | $P_7 = \frac{53.3 \times 1 \times 6250}{144 \times .2865} = 8{,}075$ | 14,018 | $P_8$ |
| | | $P_8 = \frac{53.3 \times 1 \times 6552}{144 \times .173} = 14{,}018$ | | |

## Expansion

| | | | | |
|---|---|---|---|---|
| $V_9$ | 5.73 | $T_9 = T_7 \left(\frac{V_7}{V_9}\right)^{k-1}$ | 3.46 | $V_{10}$ |
| $T_9$ | 2606° | $T_9 = 6250 \times .05^{.292} = 2606$ | 2757 | $T_{10}$ |
| $P_9$ | 168 | $T_{10} = 6552 \times .05^{.289} = 2757$ | 295 | $P_{10}$ |
| | | $P_9 = \frac{53.3 \times 1 \times 2606}{144 \times 5.73} = 168$ | | |
| | | $P_{10} = \frac{53.3 \times 1 \times 2757}{144 \times 3.46} = 295$ | | |

Work of expansion:

| | | | | |
|---|---|---|---|---|
| $W_9$ | 666,171 ft-lb | $W_9 = \frac{8075 \times 144 \times .2865 - 168 \times 144 \times 5.73}{.292}$ | 699,772 ft-lb | $W_{10}$ |
| | | $W_{10} = \frac{14018 \times 144 \times .173 - 295 \times 144 \times 3.46}{.289}$ | | |

## Residual Energy

To monitor the overall accuracy of the above calculations, the

... continued over

Engine A | Residual energy continued: | Engine B

residual energy of the exhaust gases is determined, first by considering theoretical expansion to ambient pressure, then measuring the energy of the residual heat. Of the three percent fuel energy discounted earlier, two thirds will be considered energy of residual expansion, the rest of residual heat.

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $V_{11}$ | 36.11 $ft^3/lb$ | $T_{11} = T_9 (P_4/P_2)^{\frac{k-1}{k}}$ | 33.31 | $V_{12}$ |
| $T_{11}$ | 1434° | $T_{11} = 2606 \times (14.7/168)^{\frac{.325}{1.325}} = 1434$ | 1323° | $T_{12}$ |
| $P_{11}$ | 14.7 | $T_{12} = 2757 \times (14.7/295)^{\frac{.324}{1.324}} = 1323$ | 14.7 | $P_{12}$ |

$$V_{11} = \frac{53.3 \times 1 \times 1434}{144 \times 14.7} = 36.11 \ ft^3/lb$$

$$V_{12} = \frac{53.3 \times 1 \times 1323}{144 \times 14.7} = 33.31 \ ft^3/lb$$

Work of residual expansion:

$W_{11}$ 191,331 ft-lb

$$W_{11} = \frac{168 \times 144 \times 5.73 - 14.7 \times 144 \times 36.11}{.325}$$

$$W_{12} \ \frac{295 \times 144 \times 3.46 - 14.7 \times 144 \times 33.31}{.324}$$

236,019 ft-lb $W_{12}$

Residual heat energy of gas:

$$W_{13} = (T_{11} - T_1) \times C_v \times 778 \ ft\text{-}lb$$

$W_{13}$ 143,514 ft-lb

$$W_{13} = (1434 - 500) \times .1975 \times 778 =$$

$$W_{14} = (1323 - 500) \times .1955 \times 778 =$$

125,177 $W_{14}$

## Energy Count, Efficiency

Total energy of fuel is 19,800 ÷ 19 BTu or 1042.1 × 778 = 810,754 ft-lb. The

... continued over

Engine A — Energy count, efficiency cont.: — Engine B

2% and 1% discounted during combustion are 16216 and 8108 ft-lb.

| calculated, ft-lb | x accuracy factor | percent | | percent | x acc. factor | calculated, ft-lb |
|---|---|---|---|---|---|---|
| 666,171 | | | work of expansion | | | 699,772 |
| 191,775 | | | work of compression | | | 226,959 |
| 474,396 | 476,667 | 58.8 | net expansion work | 58.3 | 472,301 | 472,813 |
| 191,331 | | | residual expansion | | | 236,019 |
| 16,216 | | | add 2% | | | 16,216 |
| 26,674 | | | work of turbocharge | | | 46,701 |
| 180,873 | 181,739 | 22.4 | net residual expan. | 25.3 | 205,312 | 205,534 |
| 143,514 | | | residual heat | | | 125,177 |
| 8108 | | | add 1% | | | 8,103 |
| 151,622 | 152,348 | 18.8 | total residual heat | 16.4 | 133,141 | 133,285 |
| 806,891 | | | Total net energies | | | 811,652 |
| 810,754 | | | Total fuel energy | | | 810,754 |
| x 1.0047875 | | | Bring-to-accuracy factor | | | x .9989182 |

## Power, Torque

| Engine A | less 10% | | less 10% | Engine B |
|---|---|---|---|---|
| 476,667 ft-lb/lb air | | Net work on cylinder | | 472,301 |
| 5.73 $ft^3$ | | Volume of air per lb | | 3.46 $ft^3$ |
| .030527 $ft^3$ | | Chamber capacity | | .030527 $ft^3$ |
| .0053275 lb | | mass of air per firing | | .0088223 lb |
| 2539.44 ft-lb | | work per firing | | 4167.01 |
| 161.6 | 145.4 | HP per chamber, 35 rps | 238.7 | 265.2 |
| 404.16 | 363.7 | Torque, max. per cham. | 596.9 | 663.2 |
| 3.064 | 2.758 | HP per $in^3$ at 35 rps | 4.524 | 5.027 |
| 7.662 | 6.896 | Torque per $in^3$, max | 11.316 | 12.573 |

...cont.

| Engine A | Power, torque continued: | Engine B |
|---|---|---|
| | A 10% reduction has been made from the calculated work to capacity ratios above, since engines A&B are two-stroke units. It is not realistic to expect more than 90% charge purity at air/fuel ratios of 19:1. Further reductions to power per in³ have been made to allow for bearing and blow-by losses (which could be less than projected). It is assumed that major auxiliary systems, eg turbo, fuel pump, generator, can be powered by residual expansion of exhaust. No charge purity discount need be made for four stroke units. | |
| 2.758 | Base HP per in³ at 35 rps (2100 rpm) | 4.524 |
| 1.139 | Actual HP/in³ at 17 rps (1020 rpm), 85% base | 1.868 |
| 2.482 | HP/in³ at 35 rps (2,100 rpm), 90% base | 4.072 |
| 7.860 | HP/in³ at 105 rps (6,300 rpm), 95% base | 12.895 |
| 5.862 ft-lb | Torque at 17 rps (1020 rpm), 85% base ft-lb | 9.619 |
| 6.551 ft-lb | Torque at 105 rps (6,300 rpm), 95% base ft-lb | 10.750 |

# NOTE 6

## VARIABLE COMPRESSION RATIO, FREE PISTON CONCEPT

For a given engine having a theoretically 'free' piston, it would be useful to determine at what point the piston would take off, i.e. start to travel beyond its designed compression-ratio. That point must surely be when the work of expansion on one face of the piston together with its kenetic energy is equal to the work required for the other face of the piston to complete the designed compression. This situation will occur at a certain engine speed; beyond that speed the excess kinetic energy of the piston will cause the gas to be compressed beyond the design ratio.

Considering a two-stroke engine, the piston is powered and accelerated down the cylinder by the forces of expansion until the piston reaches the exhaust ports (considering the embodiment of Fig. $\overset{247}{[249]}$). At that point the pressure built up by expansion is no longer directed on the piston but instead to the exhaust gas reservoir, so that now theoretically the only energy moving the piston is its kenetic energy. When this kenetic energy is equal to the work required to complete combustion, piston take-off engine speed is reached. To simplify the calculations, the little work of expansion still affecting the piston after it has exposed the exhaust ports is assumed to be balanced by the restraining effect on the piston of the tensile member, which in practice cannot vary its length without causing some expenditure of work, (except in certain embodiments where yarn is used).

. . . . continued over

So, piston take-off will start when:

Work of compression = Kenetic energy of piston

$$\frac{P_1 V_1 - P_2 V_2}{k-1} = \frac{M_p v^2}{2g} \qquad (a)$$

Where: $P_1, P_2; V_1 V_2$ are initial and final volumes & pressures, lb/in
$M_p, M_a$ are masses of reciprocating parts and charge gas
$v$ is velocity in ft/sec, $g$ is gravitational accelation, ft/sec
$C_r$ is compression ratio, $T_1$ initial temperature °R
$k$ is the specific heat ratio $C_p/C_v$

Using adiabatic relationship $P_1 V_1^k = P_2 V_2^k$ and rearranging:

$$\frac{P_2 V_2}{P_1 V_1} = \frac{V_1^{k-1}}{V_2^{k-1}} = C_r^{k-1} \qquad (b)$$

Factorizing equation (a):

$$\left(\frac{P_1 V_1}{k-1}\right)\left(1 - \frac{P_2 V_2}{P_1 V_1}\right) = \frac{M_p v^2}{2g}$$

Substituting from equation (b):

$$\left(\frac{P_1 V_1}{k-1}\right)\left(1 - C_r^{k-1}\right) = \frac{M_p v^2}{2g}$$

But $PV = M_a \times 53.3 \times T_1$ (Ideal Gas Laws), so substituting:

$$\frac{M_a \times 53.3 \times T_1}{k-1}\left(1 - C_r^{k-1}\right) = \frac{M_p v^2}{2g}$$

$$v^2 = \frac{53.3 \times T_1 \times (1 - C_r^{k-1}) \times 2g \times M_a}{(k-1) \times M_p}$$

Let mass of piston divided by mass of charge gas be the mass ratio $M_r$, and converting negative value to positive by transposing the values of the term $(1 - C_r^{k-1})$ and $g = 32.2$ ft/sec

$$v = \sqrt{\frac{53.3 \times T_1 \times (C_r^{k-1} - 1) \times 2 \times 32.2}{(k-1) \times M_r}} \qquad (c)$$

Assuming that in the case of 2 stroke engines A & B (see notes 5 and 7) the exhaust ports are exposed at 65° bef BDC crank angle, then at that point the compression requi

..cont.

to achieve the design compression ratio (20:1) will be 3.7:1. That effected already is 5.4:1. With a 4" stroke, at 45° BDC the rate of piston travel is $\pi \times .33\dot{3} \times \cos 45°$ ft per revolution which is .7405 ft per revolution, or 1.3505 revolutions per foot travel. Therefore, from equation (c) we get:

$$rps = 1.3505 \times \sqrt{\frac{3432.5 \times T_1 \times (C_r{}^{k-1} - 1)}{(k-1) \times M_r}} \qquad (d)$$

The weight of the relevent reciprocating parts (piston, tensile or compressive element, bearings) per in³ displacement is about 0.1 lb/in³ conventional engines, based on data for the Volvo TD100 B. Assume that the overall weight reduction attainable by substituting ceramics for metals and by using tensile links rather than connecting rods is 45%, then reciprocating mass per in³ in engines A & B will be 0.055 lb. Discounting 10% for those portions of the link that effectively revolve, adding about 20% for greater engine performance, discounting about 15% because loads in a two stroke twin combustion chamber cylinder are not cummulative on the crank (only net loads are transfered), then mass/in³ is .0505 lb. For the 52.75 in³ engine A combustion chamber the effective reciprocating mass will be 2.66 lb, say 2.7 lb. Because the similar sized engine B is running at higher boost pressure, components will need to be strengthened to mass say 4.0 lb. The charge masses per firing for engines A & B are given as .004742 lb and .007356 lb respectively (see notes 5 and 7) so $M_r$ values are 569.4 and 543.8 respectively. After compression to 5.4:1, $T_1$ will be $767 \times 5.4^{.36} = 1408$ °R and $990 \times 5.4^{.343} = 1765$ °R respectively. In considering the work of remaining compession, mean temperatures are estimated to be 1924° and 2212° respectively, with corresponding values of $k$ of 1.327 and 1.321.

...continued over

Using the values obtained above and substituting in equation (d), piston take-off speeds for engines A & B are:

$$\text{Engine A} = 1.3505 \times \sqrt{\frac{3432.5 \times 1924 \times (3.7^{.327} - 1)}{.327 \times 569.4}} = 186 \text{ rps } (11{,}151 \text{ rpm}$$

$$\text{Engine B} = 1.3505 \times \sqrt{\frac{3432.5 \times 2212 \times (3.7^{.321} - 1)}{.321 \times 543.8}} = 203 \text{ rps } (12{,}209 \text{ rpm}$$

As these engines are run beyond this speed effective compression ratio will increase. If they increase to 100:1, the remaining compression required after the exhaust port is exposed will be 18.5: Assuming the components can stand the increased loads, and assuming the tensile links have little restraining effect, then at 100:1 compression ratio the engines will be running at the following speeds, assuming new mean temperatures of 2402° and 2985°R with corresponding values of k at 1.317 and 1.308

$$\text{Engine A} = 1.3505 \times \sqrt{\frac{3432.5 \times 2402 \times (18.5^{.317} - 1)}{.317 \times 569.4}} = 356 \text{ rps } (21{,}363 \text{ rpm}$$

$$\text{Engine B} = 1.3505 \times \sqrt{\frac{3432.5 \times 2985 \times (18.5^{.308} - 1)}{.308 \times 543.8}} = 403 \text{ rps } (24{,}180 \text{ rpm}$$

If the engines are designed to operate regularly at variable compression ratios up to 100:1 (giving overall ratios of 300:1 a 600:1 respectively), then the mass of the reciprocating parts nee to be increased substantially to allow for the greater loads, even allowing for adjustments such as timing fuel delivery, provide a more constant pressure type of combustion. Compar notes 5 and 7 it seems reasonable to increase mass 2.5 time to 6.75 lb and 10.0 lb respectively. Mr values become 1423.5 a 1359.4. Using these new values and substituting in equatio (d), piston take-off speeds in engines A & B are:

....continued ove

Engine A = $1.3505 \times \sqrt{\dfrac{3432.5 \times 1924 \times (3.7^{.327} - 1)}{.327 \times 1423.5}}$ = 117 rps (7,052 rpm)

Engine B = $1.3505 \times \sqrt{\dfrac{3432.5 \times 2212 \times (3.7^{.321} - 1)}{.321 \times 1359.4}}$ = 129 rps (7,722 rpm)

Making the same assumptions as previously, when the above engines are allowed to run to 100:1 compression ratio, their speeds will be:

Engine A = $1.3505 \times \sqrt{\dfrac{3432.5 \times 2402 \times (18.5^{.317} - 1)}{.317 \times 1423.5}}$ = 225 rps (13,511 rpm)

Engine B = $1.3505 \times \sqrt{\dfrac{3432.5 \times 2985 \times (18.5^{.308} - 1)}{.308 \times 1359.4}}$ = 255 rps (15,297 rpm)

As will be gathered from the various constructional details described, the pistons will be at least lightly restrained by the link mechanisms. In the case of variable compression/'free' piston engines the crank links will almost certainly be of the combined tensile/compressive types, so that the great peak combustion chamber loads can always be simultaneously carried by two links, and for such engines special clearance spaces and fuel systems have to be provided.

For a given strength of construction, piston take-off speed is inversely proportional to stroke. Stroke can be reduced by 'oversquare' cylinder design, and/or by providing a greater number of smaller combustion chambers for a given engine capacity.

# FREE PISTON OPERATION - CR 100:1

Set out below are calculations for Engines A & B operating at 100:1 compression ratio, after piston take-off speed has been exceed (see note 6, Table 1). Unless otherwise noted, the same assumptions are made as in note 5, where particulars of combustion chamber and turbocharge are found.

| Engine A | | Engine B | |
|---|---|---|---|
| $V_{15}$ | ·0573 | ·0346 | $V_{16}$ |
| $T_{15}$ | 3079° | 3601° | $T_{16}$ |
| $P_{15}$ | 19889 | 38.522 | $P_{16}$ |
| $W_{15}$ | 390,582 ft-lb | 462,460 ft-lb | $W_{16}$ |

## Compression

Both volumes are compressed 100:1. Therefore $V_{15} = \cdot 0573$, $V_{16} = \cdot 0346$

$$T_{15} = 683 \times 100^{\cdot 327} = 3079$$

$$T_{16} = 825 \times 100^{\cdot 32} = 3601$$

$$P_{15} = \frac{53\cdot 3 \times 1 \times 3079}{144 \times \cdot 0573} = 19.889$$

$$P_{16} = \frac{53\cdot 3 \times 1 \times 3601}{144 \times \cdot 0346} = 38,522$$

Work of compression:

$$W_{15} = \frac{19889 \times 144 \times \cdot 0573 - 44\cdot 1 \times 144 \times 5\cdot 73}{\cdot 327}$$

$$W_{16} = \frac{38522 \times 144 \times \cdot 0346 - 68\cdot 2 \times 144 \times 3\cdot 46}{\cdot 32}$$

## Combustion

As in note 5, except that no fuel energy is discounted.

... continued on:

**Engine A** — **Combustion continued:** — **Engine B**

$Q = 1042.11$ BTu

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $V_{17}$ | .0573 | | .0346 | $V_{18}$ |
| $T_{17}$ | 7427° | | 7916° | $T_{18}$ |
| $P_{17}$ | 47,976 | | 80,683 | $P_{18}$ |

$$T_{17} = \frac{1042.11}{1 \times .2317} + 3079 = 7427$$

$$T_{18} = \frac{1042.11}{1 \times .2615} + 3601$$

$$P_{17} = \frac{53.3 \times 1 \times 7427}{144 \times .0573} = 47,976$$

$$P_{18} = \frac{53.3 \times 1 \times 7916}{144 \times .0346} = 80,683$$

## Expansion

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $V_{19}$ | 5.73 | | 3.46 | $V_{20}$ |
| $T_{19}$ | 1981° | | 2121° | $T_{20}$ |
| $P_{19}$ | 128 | | 227 | $P_{20}$ |

$$T_{19} = 7427 \times .01^{.287} = 1981°$$

$$T_{20} = 7916 \times .01^{.286} = 2121°$$

$$P_{19} = \frac{53.3 \times 1 \times 1981}{144 \times 5.73} = 128$$

$$P_{20} = \frac{53.3 \times 1 \times 2121}{144 \times 3.46} = 227$$

Work of expansion:

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $W_{19}$ | 1,010,988 ft-lb | | 1,079,805 | $W_{20}$ |

$$W_{19} = \frac{47965 \times 144 \times .0573 - 128 \times 144 \times 5.73}{.287}$$

$$W_{20} = \frac{80683 \times 144 \times .0346 - 227 \times 144 \times 3.46}{.286}$$

## Residual Energy

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $V_{21}$ | 28.98 $ft^3/lb$ | | 26.92 | $V_{22}$ |
| $T_{21}$ | 1151° | | 1069° | $T_{22}$ |
| $P_{21}$ | 14.7 | | 14.7 | $P_{22}$ |

$$T_{21} = 1981 \times \left(\frac{14.7}{128}\right)^{\frac{.335}{1.335}} = 1151$$

$$T_{22} = 2121 \times \left(\frac{14.7}{227}\right)^{\frac{.334}{1.334}} = 1069$$

$$V_{21} = \frac{53.3 \times 1151 \times 1}{144 \times 14.7} = 28.98\ ft^3/lb$$

$$V_{22} = \frac{53.3 \times 1069 \times 1}{144 \times 14.7} = 26.92\ ft^3/lb$$

Work of residual expansion:

| Engine A | | | Engine B | |
|---|---|---|---|---|
| $W_{21}$ | 132 151 ft-lb | | 168.013 ft-lb | $W_{22}$ |

$$W_{21} = \frac{128 \times 144 \times 5.73 - 14.7 \times 144 \times 28.98}{.335}$$

$$W_{22} = \frac{227 \times 144 \times 3.46 - 14.7 \times 144 \times 26.92}{.334}$$

... continued over

Note 7 continued



| Engine A | Residual energy continued | Engine B |
|---|---|---|
| | Residual heat energy of gas: | |
| $W_{23}$ 96,737 ft-lb | $W_{23} = 651 \times .191 \times 778 = 96737$ | 83 446 $W_{24}$ |
| | $W_{24} = 561 \times .1885 \times 778 = 83\,446$ | ft-lb |

## Energy Count, Effiency

| calc | x fac. | % | | % | x fac. | calc. |
|---|---|---|---|---|---|---|
| 620,606 | 611,457 | $\boxed{75.42}$ | Net expansion work | $\boxed{75.09}$ | 608,923 | 617,345 |
| 105,477 | 103,955 | $\boxed{12.82}$ | Net residual expans. | $\boxed{14.76}$ | 119,637 | 121,312 |
| 96,737 | 95,342 | $\boxed{11.76}$ | Total residual heat | $\boxed{10.15}$ | 82,294 | 83,446 |
| 822,620 | x .9856 | | Totals, correction factor | | x .9862 | 822,103 |

## Power, Torque

| Engine A | -10% | | -10% | Engine B |
|---|---|---|---|---|
| 3257.56 | | Work per firing, ft-lb | | 5371.52 |
| 207.3 | $\boxed{186.57}$ | HP per chamber @ 35 rps | $\boxed{307.64}$ | 341.82 |
| 518.45 | $\boxed{466.61}$ | Max torque per chamber | $\boxed{769.41}$ | 854.90 |
| 3.930 | $\boxed{3.537}$ | HP per $in^3$ @ 35 rps | $\boxed{5.832}$ | 6.480 |
| 9.828 | $\boxed{8.846}$ | Max torque per $in^3$, ft-lb | $\boxed{14.586}$ | 16.207 |

| Engine A | | Engine B |
|---|---|---|
| 229 rps (13,740 rpm) (max) | Speed at 100:1 CR (larger mass) (15 540 rpm) | 259 rps |
| 20.37 | HP/$in^3$ @ max speed, less add. 12% | 37.98 |
| 7.519 ft-lb | Torque/$in^3$, less add. 15% at max spd. | 12.398 |

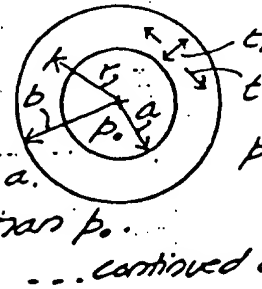
# MATERIAL STRESSES

It is apparent from Note 5 that the stresses in Engines A & B, when operated at normal compression ratios, are within the theoretical capabilities of selected high performance ceramics. However, it seems from Note 7 that it would be difficult to make the engines reliable if operated at 100:1 CR. The calculations are academic, but even so peak pressures in performance of up to 50,000 lb/in² should be allowed for. Charge pressures of ± 35000 lb/in² are produced at 100:1 CR; if one quarter of the fuel has been burnt before TDC pressures will have risen to ± 47,000 lb/in² (considering Engine B). Working to a design stress limit of 50,000 lb/in², this is likely to be approached during a very small portion of cycle time and stroke movement. In a normally compressed 6:1 boost and 100:1 engine, CR over 50:1 occurs 23.5° of cycle rotation, for 1 percent of stroke dimension. For 20:1 CR the figures are 46° and 4 percent. (The angles include both sides of TDC; peak pressures will usually occur on one side only.) The variable links will lengthen these periods

If it is assumed that special ceramics retain compressive strengths of 50,000 lb/in² and tensile strengths of 20,000 lb/in at mean operating temperatures of 4000°R, then the cylinder head assemblies of Figs. 236 [238], 237 [239] and 254 [257], 255 [258] could be reliab built, provided the units were assembled pre-stressed in compress. The relationships for stresses, $t$, in an endless cylinder is given by:

$$t_r = \frac{a^2 p_0}{b^2 - a^2}\left(1 - \frac{b^2}{r^2}\right) - \frac{b^2 p_1}{b^2 - a^2}\left(1 - \frac{a^2}{r^2}\right)$$

$$t_\theta = \frac{a^2 p_0}{b^2 - a^2}\left(1 + \frac{b^2}{r^2}\right) - \frac{b^2 p_1}{b^2 - a^2}\left(1 + \frac{a^2}{r^2}\right)$$




In both cases loads are greatest as $r$ approaches $a$.
$t_r$ is always in compression and never greater than $p_0$.

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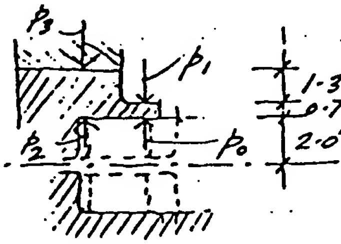
Pressure ($p_0, p_1$) in lb/in² is scalar, compression negative, tension positive. When $r = a$, and considering $t_0$, then:

$$t_0 = \frac{a^2 p_0}{b^2-a^2}\left(1+\frac{b^2}{a^2}\right) - \frac{b^2 p_1}{b^2-a^2}\left(1+\frac{a^2}{a^2}\right)$$

$$= \frac{a^2 p_0}{b^2-a^2} \times \frac{a^2+b^2}{a^2} - \frac{b^2 p_1}{b^2-a^2} \times 2$$

$$= \frac{p_0(a^2+b^2) - 2b^2 p_1}{b^2-a^2} \quad \text{and:}$$




$$p_1 = \frac{p_0(a^2+b^2) - t_0(b^2-a^2)}{2b^2}$$

$$p_0 = \frac{t_0(b^2-a^2) + 2b^2 p_1}{a^2+b^2}$$

The possible dimensions of Engines A & B are summarized above left. If the tensile design strength of the material is 20,000 lb/in², and $p_1$ is 88·2 lb/in², then the structure can withstand internal pressure:

$$p_0 = \frac{20{,}000\,(2{\cdot}7^2 - 2^2) + 2 \times 2{\cdot}7^2 \times 88{\cdot}2}{2{\cdot}7^2 + 2^2} = \frac{65{,}800 + 1286}{11{\cdot}29} = 5{,}942 \text{ lb/in}^2$$

If peak pressures at $p_2$ are 50,000 lb/in², and the tensile strength of the bead under pressure has component of 12,000 lb/in² perpendicular to $p_2$, then prestressed compression required at $p_3$ is

$$p_3 = \frac{(50{,}000 - 12000)(4^2+2^2) - 20{,}000(4^2-2^2)}{2 \times 4^2} = 16{,}250 \text{ lb/in}^2$$

Using the above material strengths, the piston portion of the piston/ro assembly can be built to withstand peak pressures. What about the rods? Around TDC, a significant portion of the loads on the piston will be converted to work to accelerate the reciprocating masses rapidly from zero velocity. The remaining loads will not be fully absorbed by the piston until there is resistance, until all slack is taken up. In the case of a piston of 12·6 in² crown area attached to rods of 3·0 in² cross-sectional area (one rod working in compression, the other in tension), the rods will carry maximum total loads of $3 \times 20{,}000 + 3 \times 50{,}00$ lb, or 16,667 lb per in² of working piston crown. If all loads were bein

...continued over

taken by the rods, this would occur when CR was about 30:1, or at 4% of stroke, or around 20° angle past TDC. If the last remaining slack is being taken up just past the above point, and in taking up slack surplus piston energies have been stored, then the cylinder module of Engine B in Note 7 is viable. As has been indicated in the disclosure, there are a number of approaches to designing bearings and crankshafts capable of transfering high loads.

Fig. 222, which is digramatic and therefore not to scale, illustrates on method of absorbing peak loads in a piston, before transfer to rods. The compressible element may be any type of spring, preferably one capable of returning stored energy at medium pressure. The return of such stored energy, preferably about half-way down the stroke, can be converted into useful work on the crankshaft. The various link bearing assemblies can alternatively/additionally serve as shock absorbers and energy storage devices.

It would seem that, with very circumspect design, it is just possible to build enduring engines using today's materials which can operate in the 300:1 to 600:1 overal compression ratio range.